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ON THE USE OF MODELS IN INVESTIGATING THE RESISTANCE OF SHIP STRUCTURES TO EXTERNAL EXPLOSIONS.*

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I. INTRODUCTION.

The behavior of a mechanical structure of given size and shape, under given circumstances, depends upon various physical properties of the structural materials and of the surrounding medium, as well as on the outside actions on the structure due to the circumstances in which it is placed. And when all of these properties and actions which are of any sensible importance are taken into account, it is clear that there must be some definite relation between them and the behavior of the structure, so that when they are all given, the behavior of the structure is thereby determined.

*When the following discussion was undertaken, in March, 1920, the writer did not know of Col. A. Guidoni's article on the same subject in the *Rivista Marittima* for 1910, an abstract of which appeared in the issue of this JOURNAL for May, 1910. Since the method here used and the points particularly emphasized are somewhat different from those in Col. Guidoni's paper, this new version may be of some interest, in spite of its lateness.

If only the shape of the structure is given, its size is one of the possible variables, and it may be specified by the value of some selected linear dimension D. Let P, P',—etc. represent the physical properties in question, such as the tensile strength of the structural material or the density of the surrounding air or water. Let A, A', etc. represent the outside actions on the structure, e.g. forces, couples, etc. Let X, X', etc. represent quantities which define the behavior of the structure,—accelerations, deformations, etc. Then the fact that the behavior of the structure depends on and is completely determined by the properties P, the outside actions A, and the absolute size D, may be expressed by saying that for each X there is an equation

$$X = F\ (D,\ P,\ P',—A,\ A',—) \qquad (1)$$

so that if the form of the operator F were known, it would be possible to find out from the equation how X would be affected by changes in any of the quantities in the second member.

A model differs from its original, geometrically, only in size, *i.e.* by having a different value of D. If the properties P are the same for the model as for the original, the model will not, in general, behave like the original when subjected to the same outside actions. For if all the P's and A's in equation (1) are kept constant, a change of D caused by substituting the model for the original will, in general, cause a change in X.

But in passing from the original to the model, it may be possible, while leaving some of the P's and A's unaltered, to adjust the others in such a way that each X shall remain unchanged, notwithstanding the diminution of D. If this is done, the behavior of the model under these new conditions will be the same as that of the full sized structure under the original conditions contemplated for it; and observations on the model will show how the original structure would behave, if the full scale experiment were carried out.

The foregoing paragraph contains the principle on which all model experiments must be conducted in order to furnish reliable information about the full scale originals. The model is made geometrically similar to the original, with its size diminished in some known ratio. It usually happens that some of the physical properties, P, can not be adjusted arbitrarily. With a ship model, for example, it may not be practicable to use any other liquid than water, so that if the density of the medium is one of the P's which affect what happens, this P has to remain fixed. Some of the outside actions may also have to be left unaltered,—gravity for example. But if there remains a sufficient number of the P's and A's which can be controlled and given arbitrarily chosen values, the model may be made to behave like the original. The problem of conducting valid model experiments consists, therefore, first in finding out what these adjustments must be, and second in selecting the materials of construction of the model and arranging the circumstances of the model experiment so that the requirements shall be satisfied.

The first part of the problem is purely theoretical, and its solution tells us what ought to be done. The second part consists in the carrying out of these instructions, which may or may not be practicable. But in any event and whatever the nature of the phenomenon under consideration, the first thing to be done when considering the possibility of getting useful information from experiments on models, is to investigate the conditions which must be satisfied in order to make the model experiment "correspond" to the full scale one, so that the behavior of the model shall be "dynamically similar" to that of the original.

The particular mechanical system now to be considered is a ship structure of given design, size, and material, which is subject to deformation by the explosion of a mine, torpedo, depth charge, or aerial bomb. If the model is

built of suitable material and all the essential circumstances of the model explosion are properly arranged, the model will be deformed in just the same way as the larger structure, and it will be possible to determine from models, not only the relative strength and effectiveness of different types of construction, but also the absolute amount of injury that would be caused to the full sized structure by any sort of explosion that can be imitated on the small scale. We have to consider what the requirements are for this sort of model experiment, and how they may be satisfied in practice.

2. GEOMETRICAL SIMILARITY.

The first requirement of all is that in the initial state, before the explosion, the whole model system shall be a complete small scale reprodution of the larger one, this geometrical similarity including the shape and size of the explosive charge and its container, the position of the charge with relation to the ship structure, and for submerged explosions, the depth of immersion. Parts of the structure which are so far from the seat of the explosion as not to be sensibly strained by it may, of course, have any convenient shape and construction, but in the parts directly concerned, geometrical similarity must be carefully attended to.

With built up structures, it is not practicable to make small scale models accurately similar to their large originals. But in view of the difficulty of duplicating such structures very closely, even with no change of scale, it is evident that a higher order of reproducibility can not be expected for the experimental results, and it is useless to insist on refinements in one direction which will inevitably be swamped by errors or uncertainties in another. It seems likely that the model can always be made sufficiently similar to its original for the practical purpose in hand, which is to make the strength of the model correspond to

the strength of the original. And the effect of departures from exact similarity can be tested by comparing models of slightly varied construction, without resorting to full scale experiments.

3. THE QUANTITIES ON WHICH THE DEFORMATION DEPENDS.

Supposing the condition of geometrical similarity to be sufficiently well fulfilled, we have next to enquire what other quantities than mere size are involved in and influence the phenomenon of deformation by an explosion, these quantities representing the P's and A's of equation (1). Since it is proposed to use dimensional reasoning, the list must include all the physical quantities that are of any sensible importance, even though they may have the same values in both model and full scale experiments, and we have to consider the following things.

(*a*) *The Properties of the Structural Material.*—The behavior of the structural material will or may depend on its elastic modulus, yield point stress, and ultimate strength under sudden application of load, in tension, compression, and shear. These quantities are all stresses and of the same dimensions. Hence the most convenient way to specify them is to state the value of some one and the ratios of the others to this one, these ratios being pure numbers. We shall suppose that the tensile strength, S, is given, and the ratios of the others to S will be denoted by r, r',—etc.

When the application of stress is sufficiently sudden, the density of the material also affects its behavior and must be specified: it will be denoted by ρ.

(*b*) *The Properties of the Explosive Charge.*—The effect of an explosion depends on the volume and pressure of the gas generated. For a given mass of explosive, these depend upon: (1) the chemical composition of the reaction products, which determines their volume at given pressure and temperature; (2) the heat of reaction, which influences the

temperature and so the pressure attained; (3) the suddenness of the explosion, or the time required for it, which influences not only the heat loss during the course of the reaction, but also the effective resistance or support furnished by the water in the case of a submerged explosion, or by the air when the explosion is above water.

It would be difficult to take account of all these quantities in a general manner, but if we suppose that the same explosive is used throughout, so that the nature of the gases and the heat of reaction are always the same, it will be sufficient to specify merely the total mass of explosive and the total time required for the explosion to be completed. With a slow explosive such as smokeless powder, the form of the time curve of burning would be a matter of importance and would have to be specified. But in practice we are concerned only with detonating explosives such as gun cotton or T.N.T., and for these it may be assumed that when the charges and their containers are geometrically similar to one another and the explosive is always the same, the total mass, M, and the total time, T, are sufficient.

(*c*) *The Properties of the Medium.*—The most important function of the medium in which the explosion occurs is to support the explosive in making its blow against the ship, *i.e.* to confine the gas and keep its pressure up. This inertia resistance to being suddenly blown away is measured by the density of the medium which is therefore one of the quantities to be included in our list and in the equation corresponding to (1). It will be denoted by σ. When the explosion takes place above water, the inertia of the explosion products themselves is comparable with that of the surrounding air, but this is already taken care of by specifying their mass, M.

The speed of propagation of compression or sound waves in the medium may also be of some importance, especially in the case of an underwater charge which is not immediately in contact with the ship. It depends upon the intensity

of the compression and the density and bulk modulus of the medium. Hence the bulk modulus, e, must be specified.

(*d*) *The Hydrostatic Pressure.*—If the medium is air, as when an aerial bomb explodes on deck, the static pressure will in all ordinary cases be the same inside and outside the structure attacked, so that it will have no effect on the result and need not be further considered. But in an underwater explosion the hydrostatic pressure, determined by the depth of immersion, the density of the water, and the intensity of gravity, is added to the pressure developed by the explosive; and if it is of any serious importance, g must appear in our list of quantities. We shall, however, disregard it for the following reasons.

If the plating of a steel ship were merely a water tight envelope designed only to withstand the hydrostatic pressure of still water, it would be very much thinner than it is in practice; and to produce a destructive injury by an external explosion would require only an added pressure of the same order of magnitude as the hydrostatic pressure already acting. But in reality, ships are built much more strongly than this in order to withstand the forces due to irregular or changing distribution of weight, working in a seaway, etc. The requisite strength might be provided almost entirely by the internal construction, with very light skin plating; but in practice, it is provided mainly by thickening the side, bottom, and deck plating so as to convert the ship into a box girder. The result is that the strength of the ship's side against a blow from without or a sudden application of pressure over a limited area, is vastly greater than would be needed if resistance to hydrostatic pressure were the only thing to be considered in the design; and to produce a destructive injury requires the application of an external pressure many times greater than the hydrostatic pressure.

Under these circumstances, the magnitude of the hydrostatic pressure must be a matter of very little importance,

and the value of g may be dismissed from further consideration without seriously impairing the value of the conclusions to be drawn.

The foregoing argument evidently applies only to surface vessels. With submarines, resistance to hydrostatic pressure is a primary consideration, and when the submarine is deeply immersed, her reserve strength available for withstanding the effect of a depth charge or mine, is very much less than when near the surface. In considering how model experiments on the strengths of submarines should be conducted, the intensity of gravity has to come in as one of the essential factors, just as it does in deducing "Froude's law of Comparison" for the resistance due to gravitational surface waves. The following deductions are therefore concerned only with surface vessels and not with deeply immersed submarines.

The quantities enumerated may now be tabulated, with their symbols and dimensions, as follows:

Description of the quantity	*Symbols*	*Dimensions*
A linear dimension of the structure	D	l
Tensile strength of the structural material	S	$m\,l^{-1}t^{-2}$
Ratios of the other strength constants to S	r, r' etc.	1
Density of the structural material	μ	$m\,l^{-3}$
Mass of the explosive charge	M	m
Time required for complete explosion	T	t
Density of the medium (water or air)	σ	$m\,l^{-3}$
Bulk modulus of the medium	e	$m\,l^{-1}t^{-2}$

4. THE FUNDAMENTAL EQUATION.

Having made up the foregoing list of quantities after mature consideration of the Physics of the case, we now *assume* that for geometrically similar structures and for

geometrically similar charges of the same sort of explosive, similarly placed and fired at corresponding points, the values of these quantities are sufficient to determine what sort of deformation shall be produced by the explosion. All the rest of our work is based on this assumption, and if we have overlooked any essential point and so omitted some important quantity, we shall arrive at a false conclusion. We have intentionally disregarded certain things which we know must have some influence,—for example, the viscosity of the water, to say nothing of gravity which has been considered in detail. But in view of the inevitable variations in the models and the very moderate precision to be expected in the experimental results, it appears that we have taken account of everything that is at all likely to have any serious effect, so we adopt the assumption and proceed to formulate it algebraically.

A deformation may be described by stating the values of a certain number of lengths L, L′, etc. For example, we may locate a number of points on the surface of the undeformed structure and, after the explosion, measure the distance L from the initial to the final position of each point. The shape of the deformation in relation to the whole structure is described by the ratios $L/D = X$, $L'/D = X'$,—etc., and the X's together with D specify both the size of the structure and the relative amount and nature of the deformation or injury produced. Our fundamental assumption amounts, therefore, to saying that for each of the X's there is some equation.

$$X = F(D, S, r, r', \ldots p, M, T, \sigma, e) \quad (2)$$

which connects the relative displacement X with the other quantities. This is the particular form which equation (1) assumes in the present problem.

Since S and e are both stresses, it is convenient to suppose only one of them to be given directly while the other is represented by its ratio to the first; and a similar remark

applies to the densities ρ and σ. The equation may therefore be rewritten in the form

$$X = F_1\left(D, S, \sigma, M, T, \frac{\rho}{\sigma}, \frac{S}{\rho}, r, r' \text{---etc.}\right) \quad (3)$$

where F_1 is an operator or functional symbol of unknown form, and the first 5 of its operands or arguments are dimensional quantities of different kinds, while the remainder, together with X, are dimensionless numerical ratios.

5. APPLICATION OF DIMENSIONAL REASONING.

The principle of dimensional homogeneity, in the form of the Π theorem*, enables us to reduce any equation, such as (3), containing 5 dimensional quantities which require 3 fundamental units, to the form

$$X = f(\Pi_1, \Pi_2, y, y', \text{--- etc.}) \quad (4)$$

in which y, y', — etc., represent such dimensionless ratios as appear in the original equation, and the (5–3) or 2 Π's are any two independent dimensionless products that can be formed by multiplying together powers of the original 5 dimensional quantities. The form of the new operator, f, remains unknown, but the number of independent arguments has been reduced from 5 to 2.

Various pairs of Π's may be found, but for our present purpose a convenient pair is $\Pi_1 = M/\sigma D^3$ and $\Pi_2 = T^2S/D^2\sigma$. These will be seen to be dimensionless upon referring to the table of dimensions given above, and they are "independent" because each involves a quantity which is absent from the other. Dimensional homogeneity requires, therefore, that whatever be the form of F_1 in equation (3), the equation must be reducible to the form

$$X = f\left(\frac{M}{D^3\sigma}, \frac{T^2S}{D^2\sigma}, \frac{\rho}{\sigma}, \frac{S}{\rho}, r, r' \text{---}\right) \quad (5)$$

*For a discussion of this theorem with illustrative examples, the reader who is not already familiar with the subject may refer to *Trans. Am. Soc. Mech. Eng.*, vol. 37, p. 263, 1915.

there being one equation of this sort for each of the X's which describe the shape of the deformation produced by the explosion.

Dimensional reasoning does not tell us anything about the forms of the operators, f, and they remain undetermined. But the fact that equation (3) can be reduced to the form (5) shows that the shape of the deformation is not determined by the values of D, S, σ, M, T independently, but by the values of the two combinations $M/D^3\sigma$ and $T^2S/D^5\sigma$, together with those of ρ/σ, S/e, r, r', etc., which have been left unaffected. In other words, the application of the principle of dimensional homogeneity reduces the number of independent variables on which each X depends, by the number of fundamental units, or 3 in the present instance.

6. THE CONDITIONS TO BE SATISFIED IN MODEL EXPERIMENTS.

The forms of the various f's in equations (5) can be determined only by an exhaustive experimental investigation for each kind of structure, and so long as they remain unknown it is evidently impossible to compute the effects on the X's of varying any or all of the quantities which appear in the second member. But without concerning ourselves at all with the precise forms of the operators, f, we are sure, from the mere existence of equations (5), that if all the operands $M/D^3\sigma$, etc., are held constant, the X's must also be constant, and the deformation described by the X's must remain of the same shape and of the same size in relation to the structure acted on. This remark gives the key to the problem of making the model explosion "dynamically similar" to the full scale one so that observations on the model shall permit of drawing correct conclusions regarding the original. In substituting a small model for the larger structure, we diminish the value of D in a known ratio, and the problem is then to

adjust the values of the other quantities so that all the arguments of the f's shall remain unchanged.

In the first place, we may discard as impracticable the idea of using any other medium than water or air, and treat the density, σ, and bulk modulus, e, as fixed. In order to keep ρ/σ, S/e, and the r's all constant, we must then keep the density, ρ, the tensile strength, S, and all the other elastic and strength constants which enter into the r's, the same for the model as for the original. Hence if the medium is left unchanged, the model must be built of a material which has the same density, tensile strength, shear modulus, etc., as the material of which it is proposed to build the full sized structure. The obvious method of satisfying this requirement, at least approximately, is to build the model of the same steel as the original, and this approximation will probably always be close enough.

The conditions above described ensure the constancy of σ, S, ρ/σ, S/e, and the r's, and it remains only to make M/D^3 and T/D constant. The first of these requirements will be satisfied automatically if the closeness of packing is the same for the model as for the full sized charge. For it has already been stipulated that the explosive shall be the same in all cases, so that constant closeness of packing entails constancy of the apparent or "gravimetric" density. The mass is then proportional to the volume, and M/D^3 is the same for the small as for the large charge, because the dimensions of the charge are proportional to D.

Finally, the time condition, T/D = constant, must be attended to, and since the suddenness of an explosion has a very great influence on the effects produced, this condition requires rather careful examination.

7. REMARKS ON THE FULFILLMENT OF THE TIME CONDITION.

The duration of the explosion is the time required for the detonation wave, initiated by the detonator at the firing point, to sweep over the whole charge to its most

distant point and explode the whole of it. But on account of geometrical similarity the firing points are always in the same relative position and the linear dimensions of the charge are proportional to D, so that the distance the wave has to travel in the charge is also proportional to D. Hence D/T is proportional to the average speed of propagation of the wave through the charge over the longest distance it has to go, and T/D will be the same for the model as for the larger charge if the mean speed of the detonation wave is the same in both. Similar location of the detonators is obviously very important, but it needs no special comment. The speed of the detonation wave, however, requires some discussion.

It appears that the detonation wave in a high explosive may be regarded as merely a special type of elastic compression wave. The wave starts the chemical reaction at each point in the explosive as it sweeps past, and so leaves behind it a changed material and a new state of affairs; but the conditions at the wave front and the mode of propagation into the undisturbed material ahead seem to be very much the same as they would be if a compression wave of the same intensity were being propagated through an inactive material like air, water, or steel.

In such a material, compression or sound waves of low intensity are propagated at the speed $v = \sqrt{E/d}$, where E is the adiabatic bulk modulus and d is the density; and variations of intensity have no appreciable influence on the speed. But waves of great intensity move faster, and as the violence of the disturbance is increased, the sharpness of the wave front and its speed of advance also increase.

The most familiar example of this is afforded by the head waves which accompany high speed projectiles.* If the nose of the projectile is not perfectly sharp but somewhat rounded or blunt, the compression wave keeps ahead

*See, for instance, the photographs of bullets by W. A. Hyde which are reproduced in U. S. Navy Ordnance Pamphlet No. 422, September, 1913; especially Figs. 17 to 20.

of it, no matter how fast the projectile is moving. If the speed of flight is raised, the wave falls in from all sides and hugs the point more closely so as to concentrate its energy over a smaller area and raise the intensity of the compression just enough to keep the speed of the wave in front of the point up to the speed of the projectile. But as the wave trails off and spreads over a larger area so that its energy per unit area is diminished, its speed soon falls to the ordinary value for sound waves in air and thereafter remains constant, the wave front being a cone of semi-vertical angle $\theta = \sin^{-1}(v/V)$, where v is the speed of sound and V that of the projectile.

This relation of speed to intensity is also well shown by A. H. Comey's experiments* on waves set up in air by the detonation of dynamite. Near the source, before the wave had had time to spread and fall off in intensity, speeds of over 5000 m/s were observed, or 15 times the ordinary speed of sound. Sound waves from electric sparks** act in the same way, and so do the compression waves set up by underwater explosions.

The following table gives approximate values for the speed of sound waves of low intensity in several substances of various densities and compressibilities. The values are for ordinary temperatures and are given in meters per second.

Substance	*Speed* m/s	*Substance*	*Speed* m/s
Air	340	Water	1450
Cork	500±	Ebonite	1600
Paraffin	1300	Tin	2500
Shellac	1320	Ivory	3000
Kerosene	1400	Steel	5000

To judge from these values, the speeds in solid explosives would probably lie between 1000 and 2000 m/s, and the

*Rep. 7th Int. Congr. of Appl. Chemistry, Section III B, pp. 28-37, London, 1909.
**A. L. Foley, Phys. Rev. vol. *16*, p. 449, Nov., 1920.

speeds observed for detonation waves along straight trains, which run up to about 6000 m/s in gun cotton and 7000 m/s in T.N.T., are evidently no greater than might be attained in neutral substances of the same density and elasticity by increasing the intensity of the disturbance.

In an ordinary substance, however intense the initial disturbance may be, its energy is limited. Hence as the wave spreads, its intensity and speed diminish, and very high speeds can occur only near the source. But in an explosive, the high pressure gas set free furnishes a continual supply of new energy at the wave front, so that as the detonation wave proceeds it does not die down but retains a constant intensity and a constant high speed which is characteristic of the explosive, packed and confined in the given manner.

If this relation obtained from the very start, the requirement that T/D should be independent of the size of the charge would be satisfied automatically. But if the initial speed is different and the characteristic constant speed does not establish itself until the wave has gone an appreciable distance, the condition will not be fulfilled for very small charges though it will, approximately, for large ones.

In straight trains of explosive, experiments show* that the speed over the first few inches from the detonator may be either higher or lower than the characteristic constant speed from there onward. A very strong detonator produces an initial disturbance which is more intense than can be maintained by the explosive itself. The initial speed is then very high; but as the wave spreads, its intensity falls to the value that can be kept up by the reaction in the explosive, and the speed of propagation sinks to the corresponding value. With a detonator which is barely strong enough to initiate the reaction, the initial disturbance is weak and the initial speed low. But the reaction builds up the strength of the wave to the normal value,

*See A. Marshall's "Explosives," 2nd ed. 1917, vol. 1, Chap. XXXI.

and though the speed is low at the start it increases, within a few inches, to the standard value. Detonators of suitable intermediate strength start the detonation wave at just the intensity it can maintain, and the speed of propagation is then uniform, even from the beginning.

While the published determinations of the speeds of detonation waves refer to propagation along extended trains of uniform section, there is no reason to suppose that propagation in three dimensions would differ essentially from propagation in one, though the characteristic constant speed might not be quite the same. The main difference to be expected is that the high initial speed due to a very strong detonator will fall to the normal constant value more rapidly and within a shorter distance, this distance being only a few inches, even in straight trains.

Since it may be assumed that mines, torpedoes, etc., will in practice be provided with strong rather than weak detonators, we conclude that in large charges the initial variations of the speed of the detonation wave will be of small importance, and that T/D will be nearly constant and independent of size, if care is taken to have the detonators similarly placed. For small charges, such as would be used with a 1/10 scale model, a detonator of medium strength must be employed so as to keep the speed constant from the start. The required strength might be found from experiments on straight trains.

It will not generally be possible to reduce the size of the detonator as much as that of the charge, so that the geometrical similarity can not be perfect. However, the detonator may be reduced in size by using the most violent filler available.

In concluding this section it may be remarked that the time condition is particularly important when the explosion takes place in air, and that in any model experiments which may be undertaken for studying the effects of aerial bombs, the fulfillment of the time condition should be very carefully attended to.

8. RECAPITULATION OF THE REQUIREMENTS.

The conditions to be fulfilled in model experiments on explosions may now be recapitulated as follows:

(A) The model structure is to be, as nearly as practicable, geometrically similar to the original, as regards the parts on which the explosion takes effect.

Since strength is more important than mass, the main point in selecting the structural materials is to have their strength reduced in the proper ratio, rather than to have all their dimensions reduced in exactly the same ratio as D. If, for example, the thin shell plates of the model, when made of the same steel as the full sized structure, have a considerably greater tensile strength than the thicker plates which have not been rolled so much, their thickness should be correspondingly reduced, in spite of the fact that this impairs the geometrical similarity. But since a change of thickness affects the bending strength very rapidly, it would be better, if practicable, to adopt a weaker material or perhaps to soften the plates by annealing. Such a change may alter the relative values of the other elastic and strength constants and so change the r's. But so long as any ordinary steel is used, these changes are not likely to be of much importance and the tensile strength for sudden loading will probably be a sufficient criterion for selecting the steel for the models.

The most obvious difficulty is with the fastenings, for it is hardly practicable to reproduce riveting on a greatly reduced scale; and even if it were, a simpler and cheaper process would be desirable. But here again the important thing is not the precise nature of the fastenings but their strength in relation to that of the adjacent material. If experiments on a testing machine show that a small welded joint has the same relative strength and yields in about the same way as a large riveted one, the one may be substituted for the other without regard to the fact that they do not look just alike. It must be emphasized here,

however, that the tests should be made with *sudden* loading.

(B) The model charge and its envelope are to have the same external form as the large ones, the scale ratio being the same as for the ship structures. The charges are to be similarly placed; the detonators are to be similarly situated in them; and the detonators are themselves to be similar, as nearly as may be.

(C) The explosive is to be the same and its gravimetric density the same in the model as in the full sized charge, its mass being then proportional to the cube of the linear dimensions.

(D) The detonator used for firing the model charge must be of such strength as to make the speed of the detonation wave constant from the start. The proper strength may very probably be inferred from the results of experiments on extended trains of the given explosive, but this point is important and should not be slighted. For if the time condition is not satisfied, while model experiments may give reliable data on the relative strengths of different types of construction, it will not be safe to draw conclusions from them regarding the absolute amount of damage which the full sized structures would suffer, and full scale check experiments will be necessary.

Upon looking over all these requirements, it is evident that they can not be satisfied very accurately. But it seems highly probable that the errors and uncertainties due to this lack of exactness will be no greater than the variations of strength between ships built to the same drawings and specifications or, especially, than the differences, under service conditions, between the effects of different mines, bombs, etc., of the same lot. Model experiments of the sort outlined will certainly give reliable data on relative strength; and if the conditions are carefully attended to, they will give reliable, even if only approximate, information regarding absolute strengths.

9. POSSIBLE SIMPLIFICATIONS IN THE CONDITIONS.

In the foregoing discussion all the quantities that seem at all likely to be of importance have been considered, in order to be on the safe side. But it may be that we have been over cautious and that some of the quantities are really of so little importance that they might as well have been ignored and omitted from equations (2) and (3). Any such omission would eliminate one of the arguments of f in equation (5) and so reduce by unity the number of conditions to be satisfied during the model experiments. It may therefore turn out in practice that some of the conditions which appear necessary are in fact superfluous, so that they may be disregarded without vitiating the results.

To illustrate such a possibility, we return to equation (5) or

$$X = f\left(\frac{M}{D^3\sigma}, \frac{T^2S}{D^3\sigma}, \frac{\rho}{\sigma}, \frac{S}{e}, r, r', \text{---etc.}\right) \qquad (5)$$

It seems quite probable that such variations of density as occur within the range of the ordinary available structural materials will have no appreciable effect on the behavior of the structure with regard to an external explosion. And if this is so, as long as we confine ourselves to such materials, ρ need not be included in the list of quantities, and ρ/σ will disappear from equation (5). It also seems possible that compression of the medium plays only a small part in determining the result of the explosion, especially when the charge is in direct contact with the ship. In this event, equation (2) would have been sensibly complete without the inclusion of the bulk modulus, e, and if the argument S/e is omitted from equation (5), the conclusions drawn will still be nearly correct.

If, or when, the two foregoing suppositions are substantially correct, and if σ is kept constant by using the same

medium for the model as for the full scale experiments, equation (5) reduces to the form

$$X = \varphi\left(\frac{M}{D^3}, \frac{T^2S}{D^2}, r, r', \cdots\right) \tag{6}$$

In the more general treatment of the problem, the presence of ρ/σ and S/e in equation (5) made it necessary to keep the density and elastic properties of the structural material constant if the medium was to remain unchanged; but these restrictions are now absent. We have still to keep M/D^3 constant, but instead of keeping the tensile strength and the speed of the detonation wave separately constant, we now have only to keep T^2S/D^2 constant, i. e., to make D/T proportional to $\sqrt{S}$. If, therefore, it is possible to control the speed of the detonation wave, as for example by some change in the mode of packing or the state of subdivision which does not alter the gravimetric density, it may be permissible to use a different material such as brass.

Whether such simplifications as these are legitimate in practice can not be decided a priori but only by experiment. The point has been discussed for the sake of illustrating how it sometimes happens that model experiments give correct results, even though some of the conditions which appear to be necessary for dynamical similarity are not fulfilled. When we make such simplifications without testing their legitimacy by experiment, we run the risk of neglecting something that is more important than it appears to be, and we must be prepared to find that our conclusions are not as accurate as we had hoped.

10. SUBMARINES.

In model experiments on submarines, hydrostatic pressure is an essential element of the problem and must be taken into account. This may be done either by putting g into equation (2) which will give the additional argument

$\sigma Dg/S$ in equation (5), or by putting the hydrostatic pressure, p, into (2) and p/S into (5), which then takes the form

$$X = f\left(\frac{M}{D^3\sigma}, \frac{T^2S}{D^2\sigma}, \frac{p}{S}, \frac{\rho}{\sigma}, \frac{S}{e}, r, r' \text{---etc.}\right)$$

If we stick to water and steel, we shall then have approximate dynamical similarity if the experiments are carried out not at depths proportional to D but at *equal* depths or equal hydrostatic pressures.

BUREAU OF STANDARDS, *April 26, 1921.*

ENDURANCE TEST OF FORCE FEED OILS.

By J. G. O'Neill, Associate.

If one understands the general principles of lubrication and has available accurate data of the service obtained from lubricating oils of known properties, it is possible for him to forecast the kind of service that will be obtained from other oils of varying physical and chemical properties, provided their properties are known and they are used under known conditions.

The engineering personnel of the Navy are familiar with the general principles of lubrication and the properties of the various lubricating oils used in the Naval Service. The results obtained from an endurance test of oils as given herein supplies information of the service obtained from light force feed lubricating oils when used in a force feed lubrication system. Information is also supplied as to the character of the changes which take place in these oils under severe service conditions.

In view of the fact that at various times statements have been made of the superior or inferior lubricating properties of animal oils or mineral oils made from different crudes, this article contains information as to the service obtained from the following oils:

Oil A—An asphaltic (napthene) base oil, well refined and classed as a light force feed and motor cylinder oil.

Oil P—A paraffin base oil, well refined and classed as a light force feed and motor cylinder oil.

Sperm Oil—Sperm oil obtained from the great cavity in the head of the sperm whale. Limpid, pale yellow oil. It is a non-drying fatty oil of low viscosity, and its viscosity is little affected by increase in temperature. It is highly valued as a lubricant and its cost is approximately five times that of either of the above mineral oils.

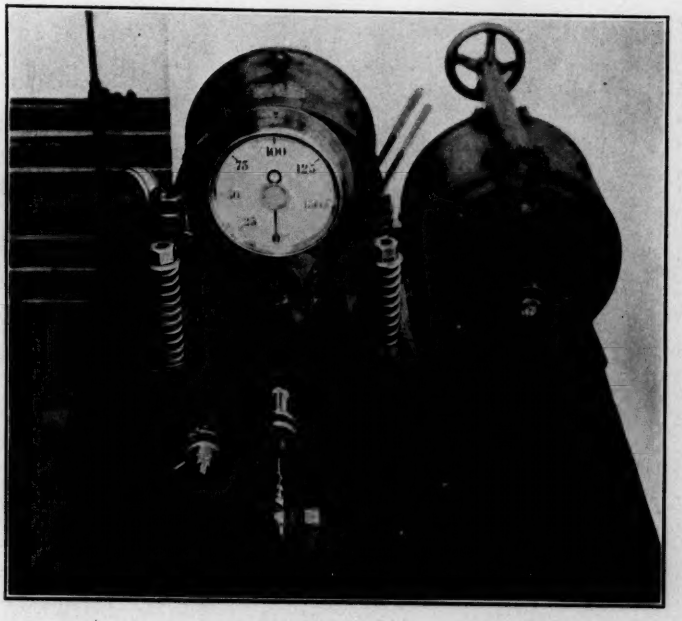
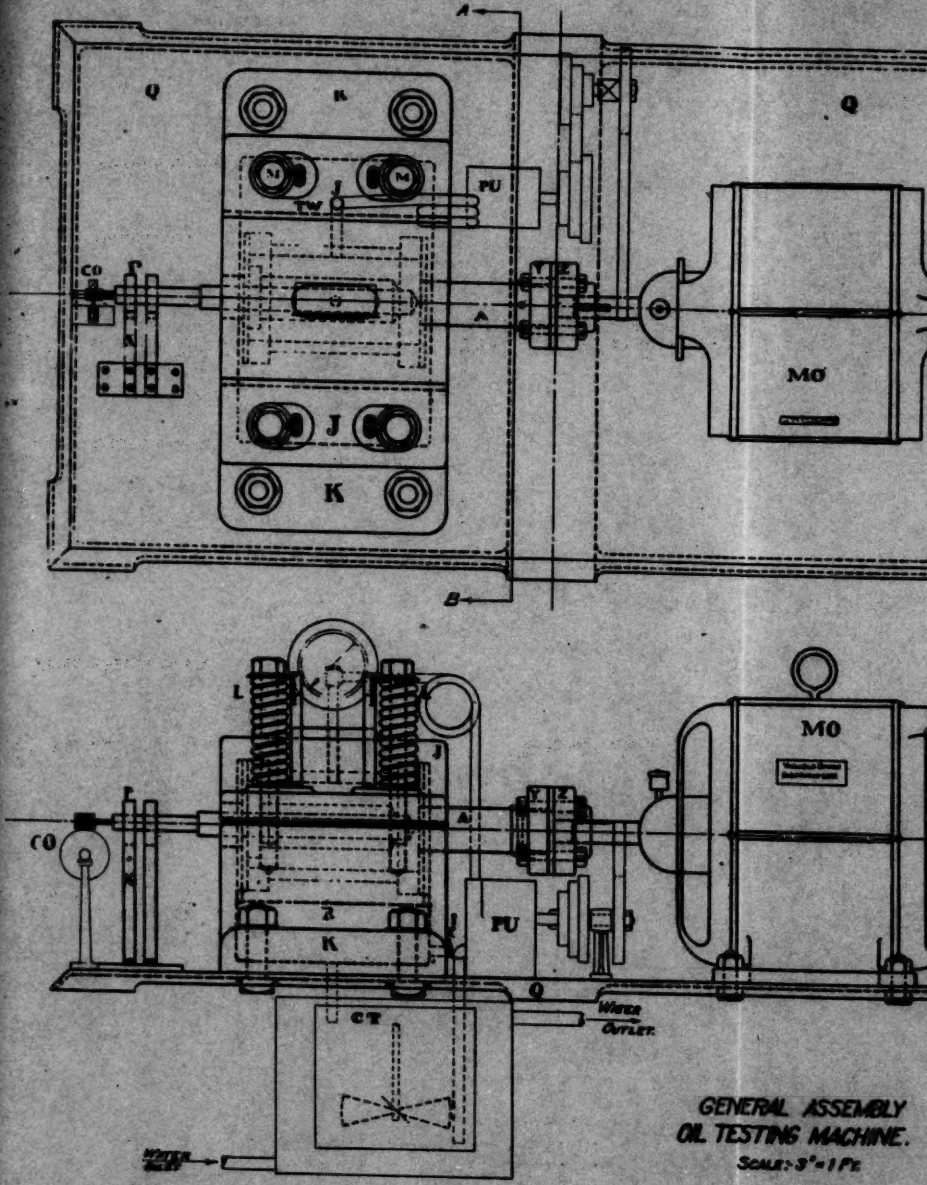
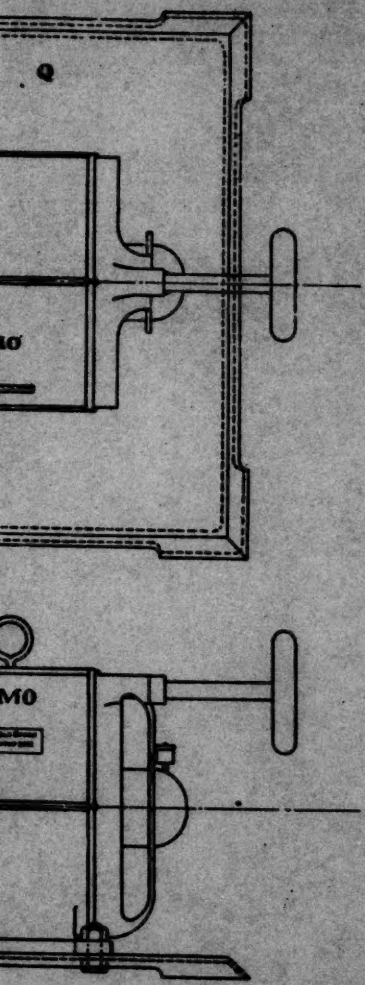


FIG. 1.



GENERAL ASSEMBLY
OIL TESTING MACHINE.
SCALE: 3" = 1 Ft



ASSEMBLY
MACHINE.
1"=1 Ft.

Fig. 2.

APPARATUS FOR TEST.

The apparatus used was an oil friction testing machine, Fig. 1, designed and built at the Engineering Experiment Station. A test journal of steel, $3\frac{13}{16}$ inches in diameter and 6 inches long, revolved in a babbitt bearing and was direct driven by a 3 horsepower variable speed motor, calibrated for foot-pounds lost work at different speeds. Force feed lubrication was used and the lubricating oil was circulated from the oil reservoir to the top on-side of the bearing by means of a motor driven rotary pump. Pressure was uniformly applied to the test journal by four calibrated springs, each provided with a scale.

As shown in Fig. 1 and Fig. 2, MO, the driving motor was connected to the test journal by a flexible coupling, YZ. Gauges showed the oil film pressure and oil circulating pressure. A counter, CO, gave a record of the revolutions made. The four springs, L, served to apply pressure to the test journal as the nuts, M, were tightened. Thermometers gave the respective temperatures of the bearing and the circulating oil. PU, the oil circulating pump was driven by a ¼ horsepower variable speed motor. X was a pair of brushes, which, if necessary, could be used to transmit current to a heating coil within the test journal by means of the slip rings, P. An oil gauge was connected to the oil reservoir to show the amount of oil it contained.

METHOD OF TEST.

For all of the runs a speed of two thousand revolutions per minute of the test journal was maintained and the circulating oil to the test journal was maintained at 15 pounds pressure per square inch, gauge.

The bearing pressures during the test of each oil were varied from 30 pounds per square inch to 150 pounds per square inch by increments of 30 pounds per square inch. A run of 30 hours was made at each bearing pressure, thus

making a total run on each oil of 150 hours. The total run of 150 hours at the different bearing pressures was made on an oil before changing to a different oil.

Samples of each oil were taken before and after being used in the friction machine and during the runs the following data were taken:

Speed of journal, revolutions per minute.
Room temperature, degrees Fahr.
Oil temperature, degrees Fahr.
Bearing temperature, degrees Fahr.
Bearing pressure, pounds per square inch.
Oil circulating pressure, pounds per square inch.
Oil film pressure, pounds per square inch.
Line voltage.
Field voltage.
Armature current.
Armature position.

From time to time sufficient oil was added to make up for leakage losses. In conducting this test no use was made of the heating coil enclosed in the test journal, since the oils used were of low viscosity and flowed freely through the bearing. It was considered as more desirable for the purpose of this test, neither to heat or cool the circulating oil, but to allow it to attain any temperature which it would during the test, since the heat generated by the oil in the bearing was a measure of the suitability of the oil.

RESULTS OF TEST.

The service obtained from the three oils is shown in Figures 3, 4 and 5. The coefficient of friction, that is, the ratio between the total friction and the total load of the bearing, was computed from an alignment chart made especially for the friction machine, Figure 6.

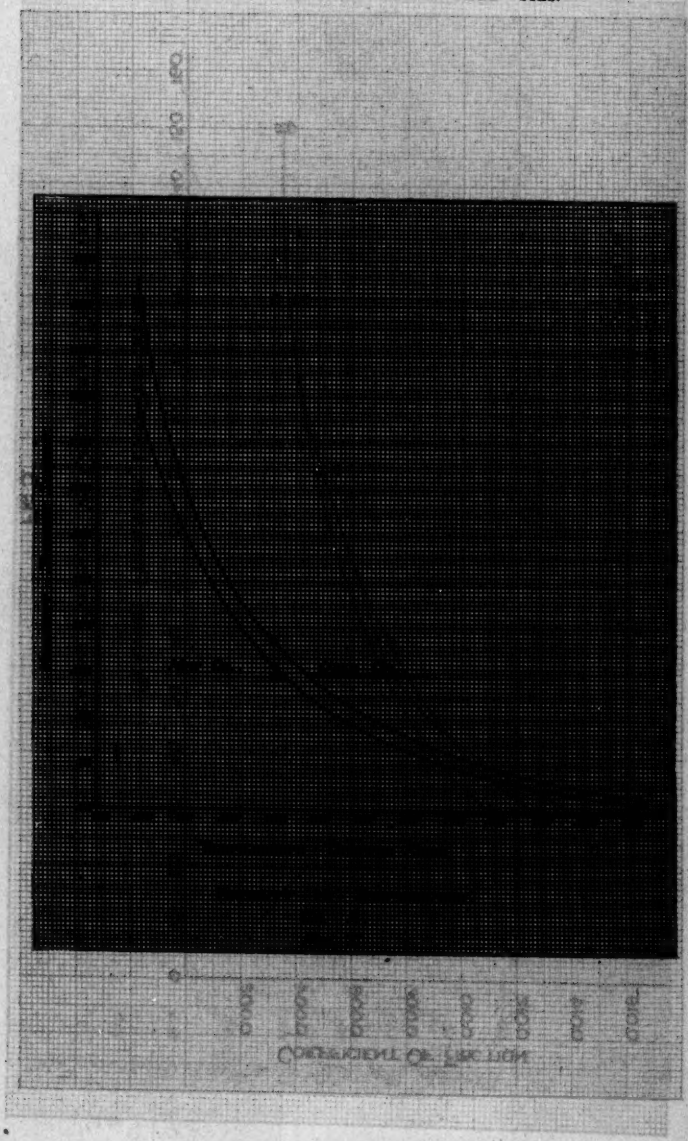
The physical and chemical changes which took place in the oils during the test are given in Figures 7, 8 and 9 and Tables I and II.

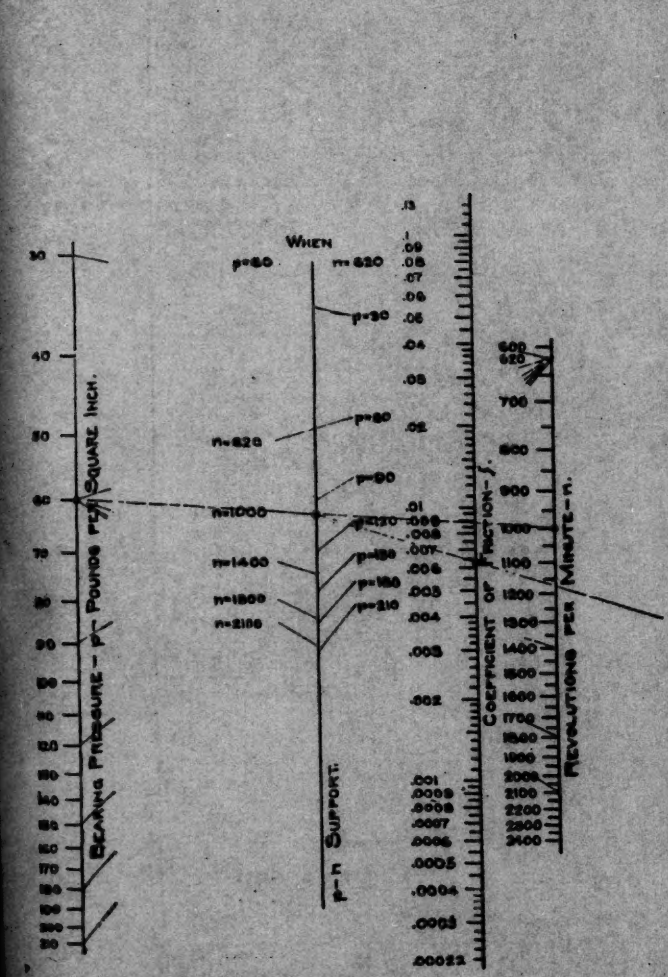
Service Data—Endurance Test Of Oils

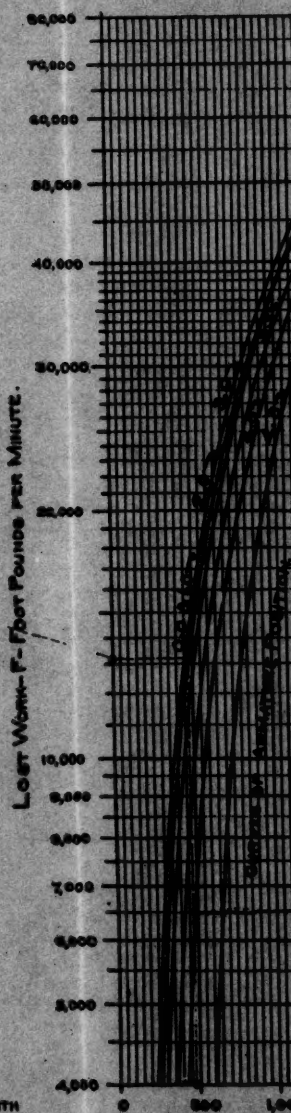
Name Of Oil	Coefficient Of Friction					30 Lb. Bearing Press.			60 Lb. Bearing Press.			90 Lb. Bearing Press.			120 Lb. Bearing Press.			150 Lb. Bearing Press.		
	At 30 Lbs Bearing Pressure	At 60 Lbs Bearing Pressure	At 90 Lbs Bearing Pressure	At 120 Lbs Bearing Pressure	At 150 Lbs Bearing Pressure	Bearing Temperature Deg. Fahr.	Room Temperature Deg. Fahr.	Difference in Room Temp. and Bearing Temp. Deg. Fahr.	Bearing Temperature Deg. Fahr.	Room Temperature Deg. Fahr.	Difference in Room Temp. and Bearing Temp. Deg. Fahr.	Bearing Temperature Deg. Fahr.	Room Temperature Deg. Fahr.	Difference in Room Temp. and Bearing Temp. Deg. Fahr.	Bearing Temperature Deg. Fahr.	Room Temperature Deg. Fahr.	Difference in Room Temp. and Bearing Temp. Deg. Fahr.	Bearing Temp. Deg. Fahr.	Room Temperature Deg. Fahr.	Difference in Room Temp. and Bearing Temp. Deg. Fahr.
	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
A	0.0115	0.0089	0.0069	0.0043	0.0035	159	78	81	169	75	94	182	82	100	195	82	113	217	86	131
P	0.0112	0.0085	0.0063	0.0040	0.0031	162	85	77	172	86	86	186	89	97	193	84	109	214	83	131
Sperm	0.0088	0.0076	0.0054	0.0032	0.0023	163	78	85	179	84	95	185	82	103	200	83	117	209	80	129

NOTE—
SIZE OF TEST JOURNAL—6 IN. LONG X 3 5/16 IN. DIA.
SPEED OF TEST JOURNAL FOR ALL RUNS WAS 2000 r.p.m.

Fig 3



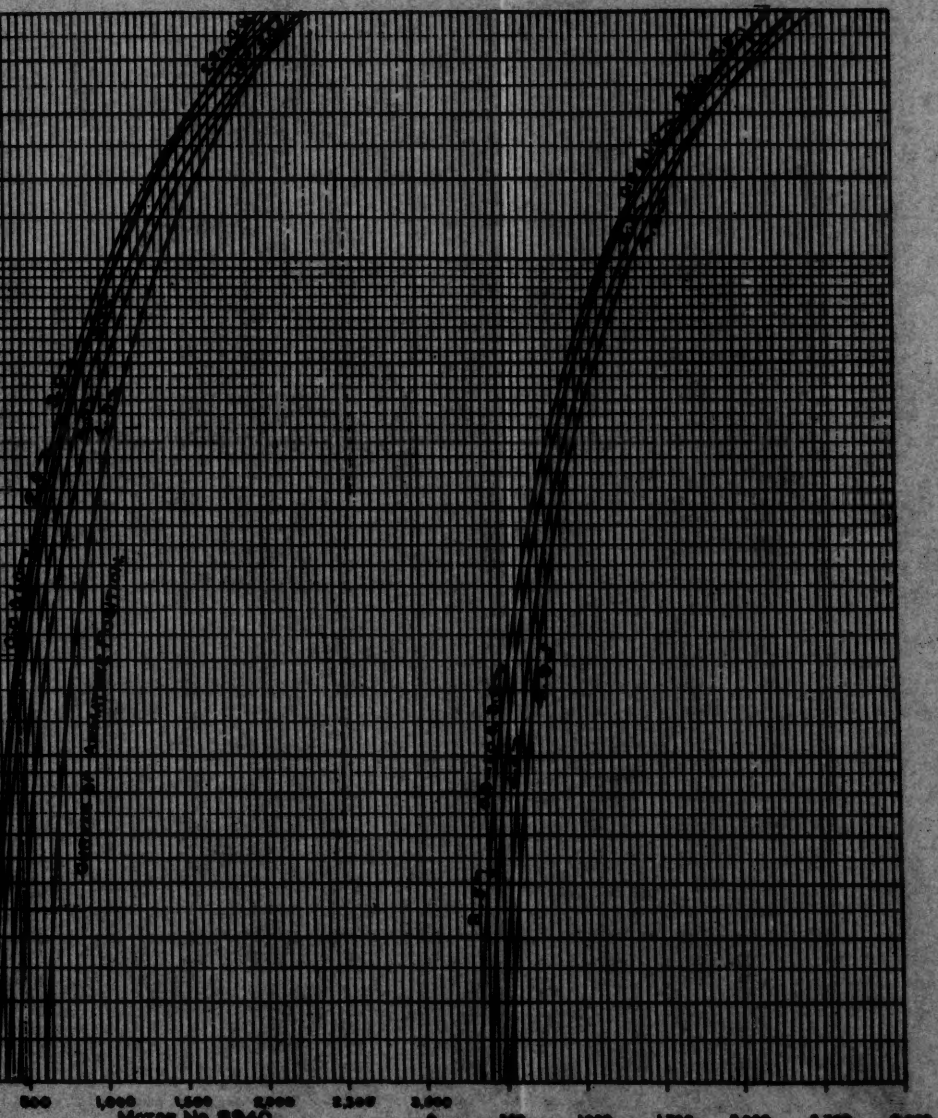




Problem:-
Pressure - 60 Pounds per Square Inch.
Speed - 1000 R.P.M.
Armature Position - 2.0.
Watts Input - 475.
Motor No. 3940.

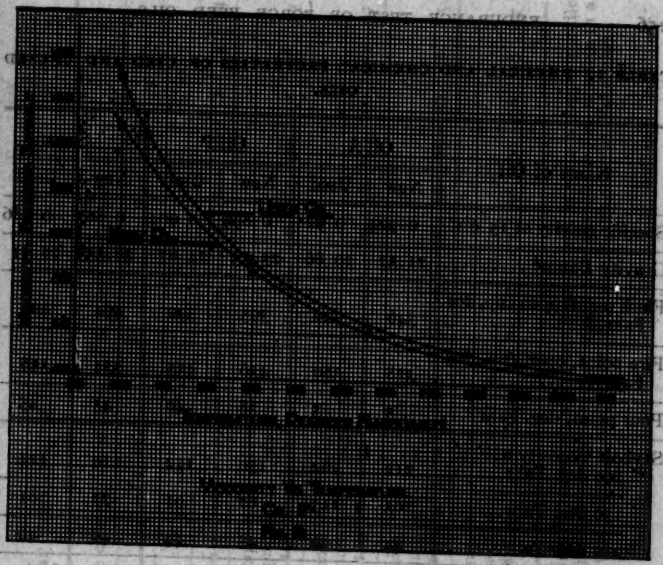
Solution:-
Join 60 on Scale p with 1000 on Scale n, and mark the intersection with the p-n Support.
From 475 Watts on the Scale for Motor No. 3940 project vertically to intersect Armature Position Curve 2.0, then horizontally to Scale F.
Join this point with the one previously marked on the p-n Support, and read the Coefficient of Friction 0.0063 on Scale f.

Fig. 6.



6.

CHART FOR FRICTION CALCULATIONS.
STATION DESIGN OIL TESTING MACHINES.



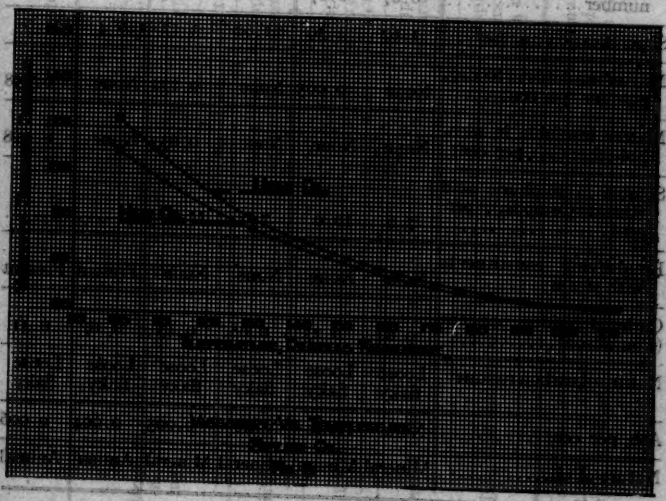


TABLE I. PHYSICAL AND CHEMICAL PROPERTIES OF USED AND UNUSED OILS.

Name of Oil	Oil A		Oil P		Sperm	
	New	Used	New	Used	New	Used
Specific gravity at 15.6°C.	0.924	0.928	0.899	0.904	0.883	0.886
Gravity Baumé	21.52	20.86	25.73	24.87	28.55	28.01
Flash point, open cup, deg. Fahr	346	340	395	390	494	490
Fire point, open cup, deg. Fahr	378	380	436	435	554	558
Pour point, deg. Fahr	5	5	38	40	45	45
Saybolt viscosity at 100 deg. Fahr	212	262	150	172	110	124
130 deg. Fahr	103	113	87	91	77	81
210 deg. Fahr	44	46	43	44	46	46
Color of oil	Pale filtered	Red	Pale filtered	Red	Pale yellow	Light brown
Organic acidity as acid number	0.07	0.67	0.06	0.84	0.84	1.40
Saponification number	1.62	4.59	1.62	3.12	148.4	129.9
Matter insoluble in 88 deg. gasoline, per cent	None	0.065	None	0.172	None	0.058
Matter insoluble in alcohol-ether (½), per cent.	None	None	None	0.053	None	0.018
Soluble in sulphuric acid (Unsat. Compds.), per cent	7.0	10.0	7.0	12.0	.....	.....
Emulsifying constituents with steam	None	None	None	None	Present	Present
Carbond residue, per cent (Conradson)	0.04	0.25	0.08	0.43	0.19	0.41
Nature of carbon residue	Loose flaky	Loose flaky	Loose flaky	Loose flaky	Loose flaky	Loose flaky
Ash, per cent	None	None	None	0.005	0.004	0.006
Nature of oil	Mineral	Mineral	Mineral	Mineral	Animal	Animal

TABLE II. EMULSION DATA OF USED AND UNUSED OILS.

Stirred at 1500 rev. per min. 40 c.c. of oil and 40 c.c. of each of the following:	Percentage of emulsion after lapse of hours indicated.											
	Oil A		Oil P		Sperm Oil							
	New	Used	New	Used	New				Used			
	½:1:4:24	½:1:4:24	½:1:4:24	½:1:4:24	½	1	4	24	½	1	4	24
Distilled water	Complete separation in 12 min.	Complete separation in 14 min.	Complete separation in 10 min.	Complete separation in 13 min.	100	100	98	94	100	98	98	97
Salt water	Complete separation in 15 min.	Complete separation in 19 min.	Complete separation in 13 min.	Complete separation in 15 min.	100	100	99	98	100	100	100	97
Normal sodium hydroxide solution	Complete separation in 10 min.	Complete separation in 9 min.	Complete separation in 16 min.	Complete separation in 10 min.	65	65	25	18	75	50	22	13
Boiling distilled water	Complete separation in 13 min.	Complete separation in 14 min.	Complete separation in 8 min.	Complete separation in 10 min.	84	78	73	58	100	98	98	97

These data were obtained by using the Station's emulsion machine. This consists of a small motor which turns a copper paddle at the rate of 1500 revolutions per minute. Equal parts of oil and each of the solutions named above were stirred for five minutes in a 100 c.c. graduated cylinder. The temperature, both while stirring and while settling out, was maintained at 130 degrees, Fahrenheit.

DISCUSSION.

The endurance test of the oils was considered as being a severe practical test for the following reasons:

(*a*) The total quantity of each oil used was only about one gallon.

(*b*) The oil was not cooled.

(*c*) The bearing pressures for the latter half of the test were higher than is generally found in lubrication systems using light force feed oils.

(*d*) The speed of the journal was 2,000 surface feet per minute, which is considered as being a fairly high speed. The high speed of the journal increased the friction and the consequent heating of the bearing.

It is interesting to note that all three oils, although they were of low viscosity, maintained satisfactory lubricating films at high temperatures, high speed and heavy bearing pressures.

In general the circulating oil temperature before entering the bearing was about 40 degrees, Fahrenheit, lower than the temperature of the bearing. The oil film pressure per square inch, gauge, showed slightly higher than the bearing pressure per square inch.

Figures 3 and 5 show that the coefficient of friction at different bearing pressures for Oil A and Oil P vary directly as the viscosities of these oils. The coefficients of friction at different bearing pressures for Sperm oil (animal oil) do not vary in the same manner as those of the mineral oils; at the low bearing pressures the coefficients of friction are higher and at the high bearing pressures are lower than those of the mineral oils.

Figures 3 and 4 show that the rise in temperature of the bearing above room temperature at different bearing pressures for Oil A and Oil P vary directly as the viscosities of these oils. Sperm oil, which has a lower viscosity at low temperatures than either Oil A or Oil P, gave a greater rise in bearing temperature.

As shown in Table I, all oils darkened during use and increased considerably in viscosity and gravity. Due to the fact that asphaltic base lubricating oils have a lower flash point and a higher evaporation loss than either paraffin base or animal lubricating oils, used Oil A showed the greatest increase in viscosity. The marked increase in viscosity of all of the used oils of this test, showed that the conditions of test were very severe. Such a marked change in the viscosity of force feed oils during use seldom takes place in less than 1,000 hours of ordinary service.

The increases in organic acidity, oxidized matter, unsaturated compounds and carbon residues of all oils were in accordance with the severity of the test, but these changes did not show extensive breaking down of any of the oils.

Straight mineral oils during use increase in saponifiable resins by oxidation and the saponification numbers of used mineral oils are a measure of their oxidation. The two used mineral oils showed that they had been considerably oxidized during the test. The saponification number of used Sperm oil evidently does not measure the oxidation of the Sperm oil, because its saponification number was lowered by use.

Table II shows that lubricating oils are not changed materially in their non-emulsifying or emulsifying properties by being subjected to severe service tests. From this table it is apparent that Sperm oil on account of its emulsifying properties would be highly unsatisfactory for use in a force feed lubrication system.

CONCLUSION.

From the results of this test it is concluded that mineral lubricating oils, derived from asphaltic base crude or paraffin base crude and of the same viscosity at operating temperatures, will give the same service in a force feed lubrication system.

Since the temperature of the oil in a force feed lubrication system generally does not exceed 130 degrees, Fahrenheit, it is concluded that Sperm oil gives inferior service with respect to coefficient of friction and heat generated, as compared to the light force feed mineral lubricating oils.

The most important change which takes place in low viscosity lubricating oils, when used in a force feed lubrication system under severe service conditions, is the increase in viscosity.

In a well designed bearing supplied with lubricating oil by means of force feed, light force feed mineral lubricating oils will maintain a satisfactory lubricating film at a bearing pressure of 150 pounds per square inch, a bearing temperature of 214 degrees, Fahrenheit, and a journal rubbing speed of 2,000 surface feet per minute.

FOUR YEARS OF BALANCING PRACTICE.

By N. W. Akimoff, Associate.

One of the writer's associates in Europe received, a few weeks ago, a letter from one of the head engineers of a large and very well known turbine building concern, which letter contains the following paragraph:

"With reference to the dynamic balance, I am satisfied, after consideration of the method of construction and manufacture of turbine rotors, and from actual experience in running a large number of rotors at the highest revolutions, that if the static balancing has been carefully carried out the residual dynamic errors of balance are negligible. My view is that in practice very small errors of balance are distributed throughout the length of the rotor, so that if the statically found balance weight is divided between the two ends of the rotor, the remaining dynamic error is so small as to give an inappreciable reaction at the bearings, and this view has been amply justified in practice."

This of course implies that the conception of dynamic balance is really something psychological and not real. The writer is quite sure that the engineer in question did not really *mean* anything of the sort (*how could he?*) but, nevertheless, he *wrote* this over his signature; he probably did not know that very often, in a rotor perfectly balanced statically, dynamic unbalance calls for corrections of the order of *tens of pounds*.

The average engineer in this country begins to understand, more or less, the difference between static and dynamic balance, and it is about time that he should do so, the subject itself being exceedingly old and well worked out in detail, at least as regards the principle. For instance in Rankine's *Millwork* (London, 1869) the principles of

both static and dynamic balance are set forth in the clearest and most elementary manner. Rankine, of course, does not suggest any particular type of apparatus on which to obtain such balance, but his exposition can naturally lead to an unlimited number of devices with which to detect the presence of an unbalanced force or of an unbalanced centrifugal couple.

However, let us review, for the sake of clearness, these two important conceptions:

Static Unbalance means that the center of gravity of a certain rotor of weight, W, is *off* the axis of rotation by a small amount, R, the measure of said static unbalance being WR in any units, say in "foot pounds," or "inch-ounces." Or, to put matters in a slightly different way, static unbalance means that a certain rotor of weight $W-w$ is in perfect balance, and, associated with it, there likewise exists a certain small weight, w, located at a certain distance, r, away from the center; the measure of unbalance being $= wr$. Of course $WR = wr$, but the latter notation possesses certain advantages over the former, involving small weights at easily measurable distances and disregarding the mass, itself, of the rotating body. Indeed, it means more to the average practical man to be told that the static unbalance of a certain rotor is 160 inch-ounces, than to be given the same information in this form: "the center of gravity of a certain rotor, weighing 10,000 lbs. is .001 inch away from the axis of rotation."

By Dynamic Unbalance is meant the condition that the principal axis does not coincide with the axis of rotation; in other words that there exists a certain "centrifugal couple;" in still simpler language this means that, although the body is in static balance, there are *two* heavy points in it, located in the same axial plane, but at a certain distance from each other, axially, and subject to the condition that each weight-times-radius is the same, otherwise there would be no static balance.

Any rotor, taken at random, is generally unbalanced both statically and dynamically. A body may be in perfect static balance but unbalanced dynamically; but to say that a body is in perfect dynamic balance, although out of balance, statically, does not really mean anything at all.

To represent matters graphically we might suggest the sketch Fig. 1, as illustrating the effect of static unbalance; the rotor is here shown simply as a bar, B, and its unbalance is represented by the small off-center weight, w. The effect is that instead of rotating quietly about the axis, A, of rotation, the body tends to rotate along the cylindrical surface, C, under the influence of the centrifugal force, developed by w.

The effect of dynamic unbalance is shown in Fig. 2. Under the actions of the two off-center weights w_1 and w_2, the body (indicated simply as a rod B) tends to describe a conical surface about the axis, A, of rotation.

Of course the bearings exert a certain reaction in both cases, for which reason both the cylinder of Fig. 1 and the double cone of Fig. 2 are of almost vanishing proportions, but these very reactions cause the vibration proper; our problem, in balancing a rotor, is precisely to eliminate these.

The conception itself, of a *balancing machine*, is possibly thirty or forty years old; however, this is really a misnomer, because no balancing machine is meant to *balance* anything, its real object being to *supply the necessary information* for calculating and introducing the corrective measures, weights or drilled holes, etc., which put the rotor in balance. From this it is easy enough to see that a great multitude of types of apparatus for indicating unbalance can be readily suggested; in fact an examination of the patent records and of old trade journals reveals a very considerable wealth of information on this subject; although most of this apparatus appears to have been proposed by purely "practical men," having no conception whatsoever of the physics, or rather of the dynamics, of the problem.

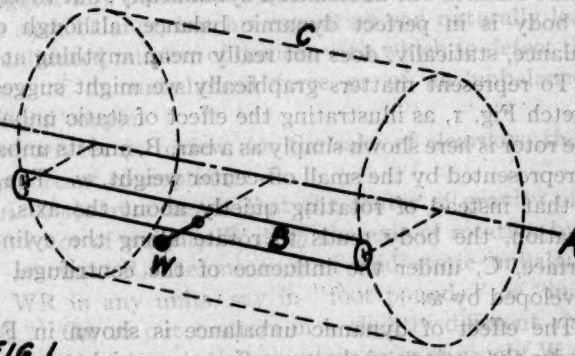


FIG. 1.

EFFECT OF STATIC UNBALANCE ON A BODY ROTATING ABOUT THE AXIS A

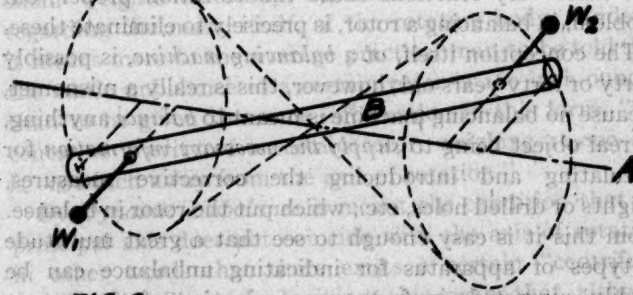


FIG. 2.

EFFECT OF DYNAMIC UNBALANCE ON A BODY ROTATING ABOUT THE AXIS A

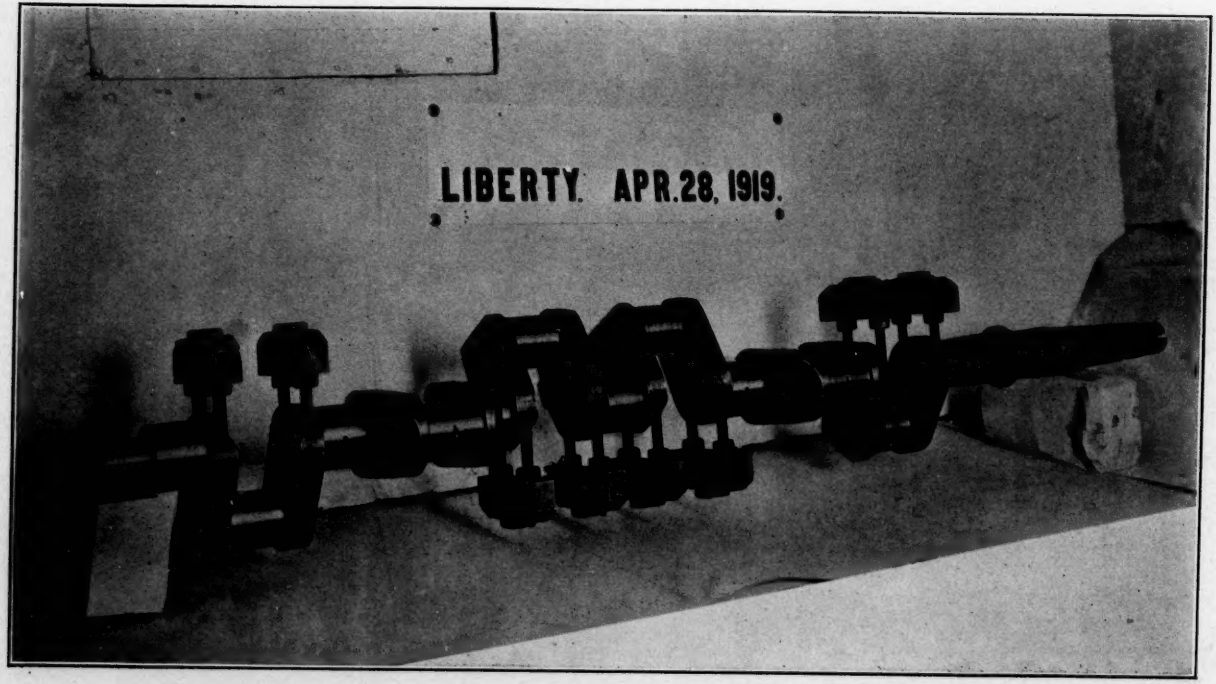


FIG. 3.

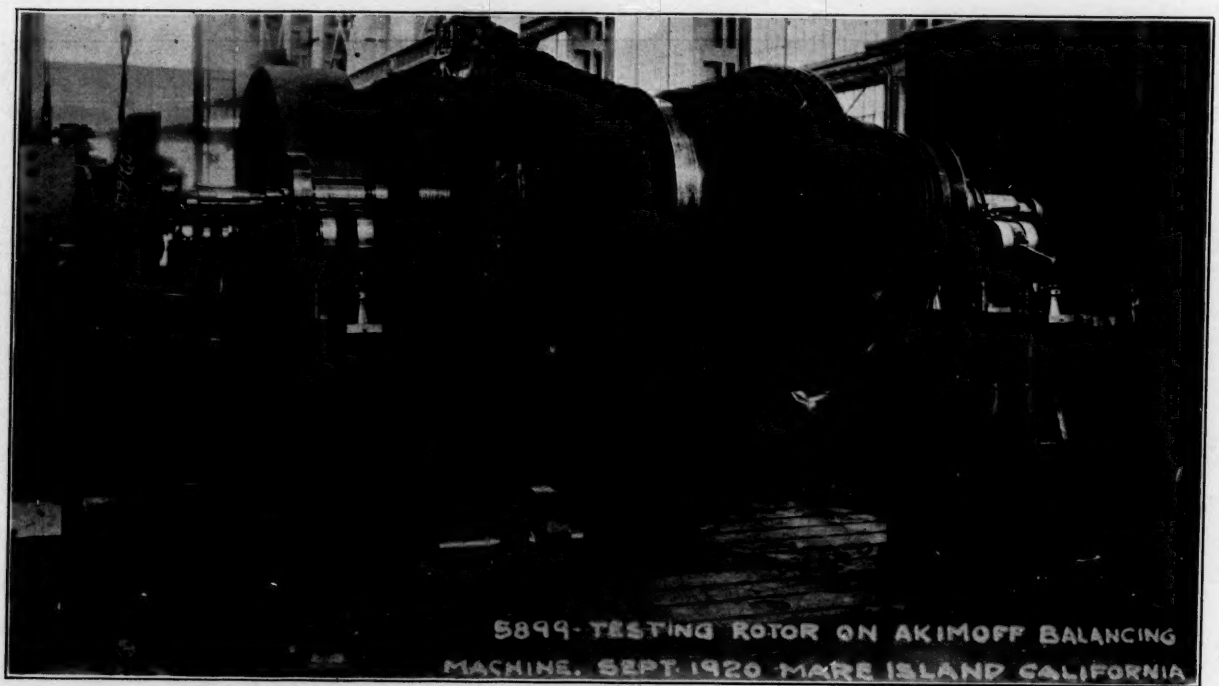

5899-TESTING ROTOR ON AKIMOFF BALANCING
MACHINE. SEPT. 1920 MARE ISLAND CALIFORNIA

FIG. 4.

FIG. 4.

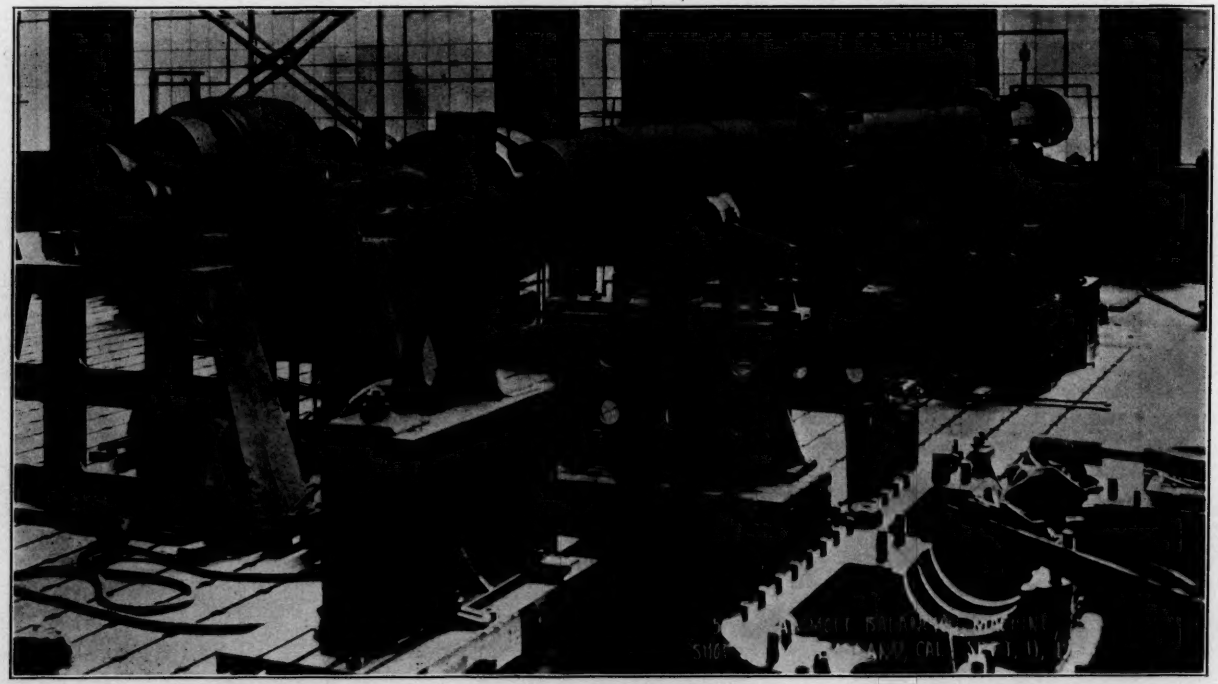


FIG. 5.

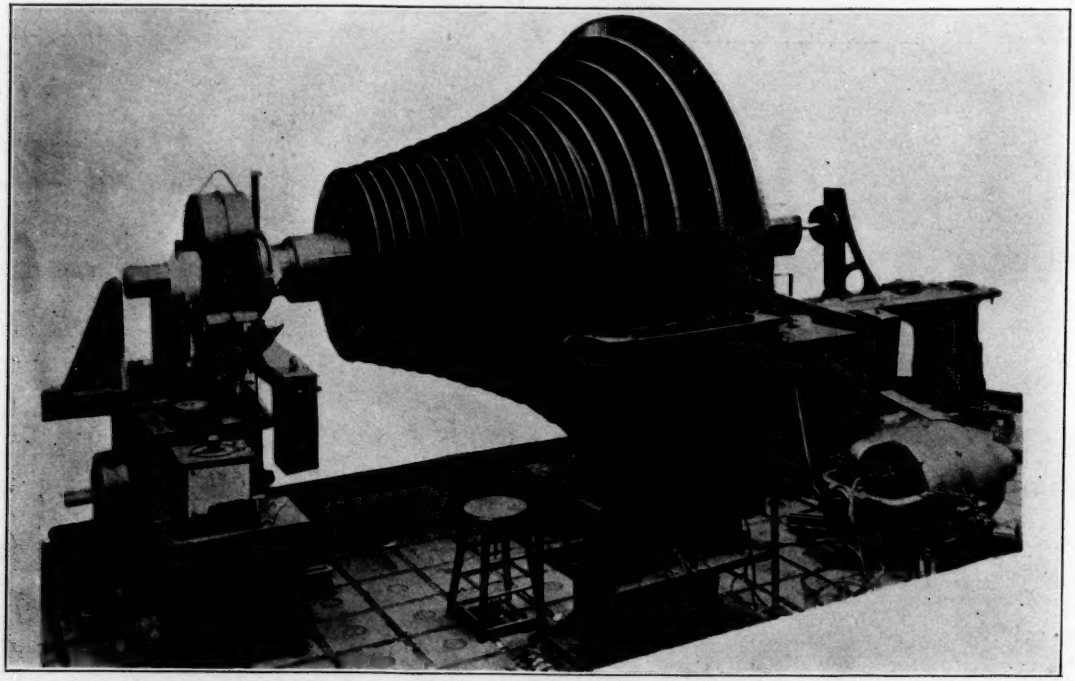


FIG. 6.

The present writer claims credit only for the rigid principle he adopted as the very foundation of his work: *physics first, construction last,* whereas the majority of designers worked on the other basis, *construction first and last and hang the physics,* whence the failures and the tremendous sums of money lost in "experimenting" in the attempt to produce "our own" balancing machine, *just as good as any.*

As a whole the writer is satisfied with the attitude of the majority of engineers and especially of the Navy; however, the three main obstacles in the development of the art are the following:

(*a*) The price of the apparatus which the average engineer invariably finds to be "much too high," even though he may never have seen the machine and knows nothing of its construction or performance.

(*b*) The usual tendency to purchase a small machine and to use it for balancing much heavier articles than the maximum agreed on in the contract; this results in the failure of the apparatus, of course, and in the usual "black eye" to the whole art, on general principles.

(*c*) The attempt to balance articles which positively are not fit to be placed on a balancing machine. Take for instance the crankshaft of a Liberty engine as shown in Fig. 3. It was designed, or rather *drawn up,* for indeed it was never really *designed,* without the counterweights. The weights shown in the figure, fastened by screws, were added in the writer's laboratory in a temporary manner and the weakness of the shaft is such that no balance could here be of any help: the terrific whipping would instantly counteract most minute balancing in an absurdly designed article of this sort.

Hence the self-evident rule for the designer of a high speed part: the design *must* be such that the article *can* be balanced, unless you want the well known results, such as the engine shaking itself to pieces, or instruments

dropping off the board; sometimes the failure of the fuel tank due to terrific vibrations; etc. Another incorrect notion is that of the speeds at which to balance: either the article is properly designed, so it does not whip, in which case it can be balanced at any speed, the lower the better: or, if the article is not well designed and is expected to deflect at higher speeds, the result of balancing, even if the machine is adapted for this, is to show the imperfection of design and the necessity of counterweights or general re-inforcing: but this can be seen from the drawing, and such an indication is *not* the object of a balancing machine. For this reason any stipulation in the specifications, such as balancing at "over-speeds" of 20 percent or so, is absolutely wrong; what is desired is the *absence of perceptible vibration at all speeds, from zero up to and including 20 percent above the maximum operative speed.*

Fig. 4 illustrates balancing of a large reduction gear, clearly seen in the background; in this connection it should be borne in mind that such gears are often noisy and balancing does not necessarily take out such noise, which might result from the profile of the teeth or mistakes in execution.

Balancing of long hollow shafts is shown in Fig. 5. The unbalance may be very considerable and usually results from the difficulty of boring the hole central with the shaft itself.

Fig. 6 shows a large turbine-rotor being balanced in the Schenectady works of the General Electric Co. The writer's object in showing this rotor is to point out the fact that a rotor of this construction, balanced cold and at a slow speed, will preserve its balance under steam and at its full speed, contrary to the assertion of many engineers.

The following definite facts have been established during the past four years:

1. It is just as easy to balance a heavy rotor as a light one.
2. The windings of electrical apparatus *do* shift; it is useless to balance any rotor where this is expected.

3. Slow speed balancing is conclusive, if the equipment is well made and properly used.

4. The best check of any balancing machine or equipment is to reverse the rotor end for end and observe whether balance previously obtained *remains balance*, as of course should be the case.

5. To use the dial gauge for indicating vibrations under high speeds is faulty practice.

6. True balance means perfect running at all speeds, unless the body is badly designed.

Balancing is now being done only from the standpoint of comfort and safety, but not necessarily from that of economy; the writer is now engaged in investigation relative to the *loss of power due to vibrations;* this work is exceedingly difficult and tedious, but the greatest research men in the country are associated with him in this work, the results of which will be made public probably this time next year. It is safe to say, however, that from all indications it appears that the findings will clearly demonstrate the very considerable loss of energy, which, under certain conditions, may result from vibrations.

BOILER FURNACE INSULATION.

By Lieut. Comdr. Frank W. Sterling, U. S. Navy, Member.

Boiler insulation may be divided into two classes, that suitable for lagging shells and pressure parts and that suitable for furnace walls and other internal locations subject to high temperatures.

The introduction of oil fuel has not only emphasized the importance of very high refractory qualities for fire brick, but has also resulted in the demand for an insulating backing of material of high insulating value that will not disintegrate under the high temperatures reached in oil fuel burning.

In coal burning boilers three general types of insulating materials found extensive use as furnace brick backing, viz: asbestos products, magnesia products and crushed brick.

The increased furnace temperatures resulting from oil burning developed frequent failures in these insulating materials. These in turn lead to tests by the Navy and by such well known companies as the Babcock & Wilcox Co. to determine a suitable material.

While magnesia and asbestos products have been developed to a high degree of efficiency for lagging pressure parts, steam pipes and other moderate temperature work, these products do not appear suitable for furnace and other very high temperature insulation:

Mr. Ernest H. Peabody states in The Marine Engineers' Handbook that, "The Babcock & Wilcox Company recently conducted elaborate experiments which showed

that 85 per cent magnesia covering, while entirely satisfactory for lagging steam drums, pipes, etc., is quite unsuitable as a material for insulating furnace walls, but that on the contrary, non-conducting materials in which diatomaceous earth is used as a basis, are satisfactory both as to lightness and non-heat-conducting qualities, as well as ability to withstand the necessary temperatures."

In one series of the above tests on 85 per cent magnesia it was found that the insulating value rapidly decreased at about 600 degrees F. and that disintegration commenced at about 1,100 degrees F.

A successful furnace insulation must have high insulating value, practically no shrinkage, must not disintegrate, all these at temperatures up to 2,000 degrees F. All else being equal, the lightest product is the most desirable. Diatomaceous block insulations of specific gravities ranging from 0.35 to 1.14 are offered by various manufacturers.

The following commercial insulations are offered for high temperature insulation.

Nonpareil, Armstrong Cork & Insulation Co.; Fire Backing, National Magnesia Co.; Zenithern High Temperature Block, The Zenithern Co.; J–M High Temperature Block, H. W. Johns-Manville Co.; Insulbrix, Quigley Furnace Specialty Co.; Celite Blocks, Sil-O-Cel Brick and Sil-O-Cel C-3, Celite Products Co.

Diatomaceous earth as mined has a stratified structure. Calcined blocks of this material would be suitable for use with but one objection. At high temperatures these blocks will fail due to the fact that fissures occur along the lines of stratification. For this reason it is usual to crush and calcine the earth and use it in lump form or mold it into blocks using some binder in the manufacture.

One such product employs an organic material and clay as a binder on the theory that during the calcining process the organic material will burn out leaving a cellular structure.

Asbestos, used in very small quantities, as a binder, promises moderate success. The writer has been told of tests with blocks manufactured by this process that showed a shrinkage of five per cent.

Shrinkage and weight must be kept to a minimum. Only approximate weights may be assigned to the various commercial blocks, as, obviously, this will depend to a certain extent upon the manufacture. Shrinkage depends upon the degree of calcining during manufacture and upon the binder used in molding blocks.

METHODS OF TESTING FURNACE INSULATION.

A comparative test of Sil-O-Cel C-3 and crushed fire brick conducted at the Fuel Oil Testing Plant, Navy Yard, Philadelphia, illustrates the installation and procedure generally followed in determining suitability of insulating material for Naval boilers.

The following is quoted from an actual test on the above two materials to determine their relative insulating value and durability for the furnace bottoms of oil burning boilers:

"*Description of Installation.*—The test of these materials was made on the furnace bottom of the Experimental White-Forster Boiler at the Fuel Oil Testing Plant in conjunction with evaporative tests. The dimensions of the bottom pan are 11 feet, 11 inches in width and 10 feet, 11¼ inches in depth.

"The right half of the bottom pan was covered with a 2-inch layer of the calcined Sil-O-Cel, Grade C-3. This material comes in lump form, the particles varying in size from a powder to those that will pass through a sieve of 4 mesh to the inch. It has a very light pink color derived from the calcining process. The raw diatomaceous earth from which it is made is pure white in color. Calcining eliminates subsequent shrinkage due to heating and cooling.

In calcining, a small percentage of the insulating value is lost but the elimination of shrinkage more than offsets this loss.

"The Sil-O-Cel, Grade C-3, was dampened a little with water and spread on the furnace pan. It was then tamped down lightly. Very little water was used, only enough to dampen, so that the material would pack well. In tamping, care was taken *not* to pulverize or break up the lumps, as to do so would considerably reduce its insulating value.

"On top of the Sil-O-Cel, Grade C-3, two layers of split brick (9 inches x 4½ inches x 1½ inches) were laid. 'Woodland' brick manufactured by Harbison and Walker Refractory Company were used. Each course was laid without cement, allowance being made for expansion on initial heating. All joints between the two courses were broken. The cracks between the brick were filled with dry high grade fire clay (Contract Refractory Cement, No. 2) and brick were then covered to an approximate depth of 1/8 inch.

"The left side of the furnace bottom pan was laid up in the same way except that crushed fire brick was used in place of the Sil-O-Cel, Grade C-3. The crushed brick was supplied by the Babcock & Wilcox Company and is the same as that used in many recent installations. [For installation see Fig. 1.]

"*Description of Test Instruments.*—The temperature of the heat passing through the bottom of the furnace was measured with a resistance type (Heraeus) platinum pyrometer, used in conjunction with an Hanovia Recorder. The accuracy of the readings over the range covered is plus or minus 2 degrees F. Two 5-foot pyrometers were used. Fig. 1 shows their location underneath the bottom pan in the air space between pan and outer casing. They were wedged up against the bottom pan.

"Furnace temperatures were taken with an improved Morse Optical Pyrometer manufactured by the Leeds and

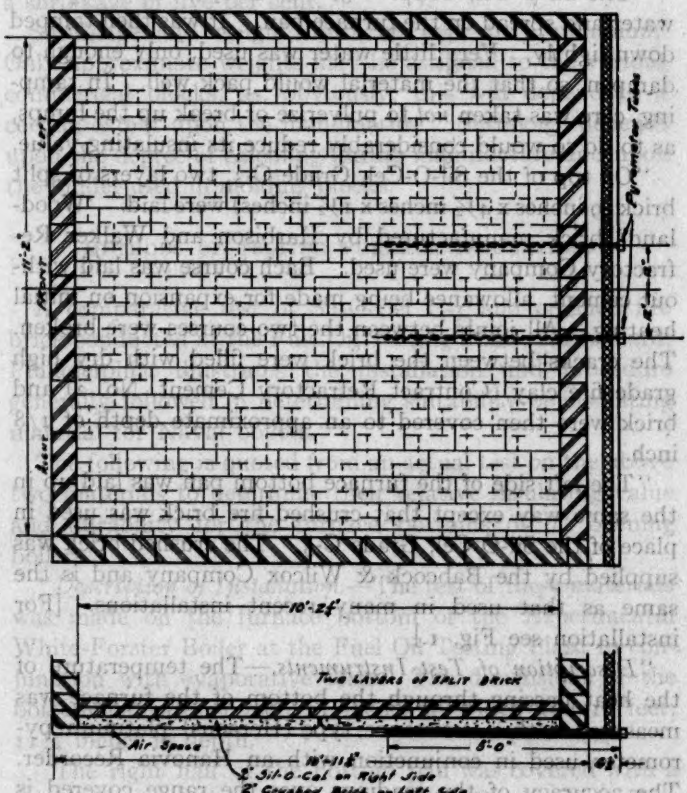

Pyrometer Tubes
11'-2"
10'-2"
Two Layers of Split Brick
Air Space
5'-0"
2" Sil-O-Cel on Right Side
2" Crushed Brick on Left Side

FIG. 1

FIG. 2. Furnace of experimental White-Forster boiler showing points of observation of furnace temperature with optical pyrometer.

NOTE: The temperatures recorded at 'A' 'B' 'C' and 'D' were temperatures observed through open sighting holes. Temperatures recorded at 'E' 'F' were under black body conditions.

TABLE 1.

Date.	Initial Temp. of Sil-O-Cel, C-3.	Initial Temp. of Crushed Brick.	Final Temp. of Sil-O-Cel, C-3.	Final Temp. of Crushed Brick.	Total Temp. rise of Sil-O-Cel, C-3.	Total Temp. rise of Crushed Brick.	Duration of run in hours.	Av. Temp. rise per hour of Sil-O-Cel, C-3.	Av. Temp. rise per hour of Crushed Brick.	Furnace Temperatures, deg. F.					Remarks
										Open hole at side "A."	Closed tube at side "B."	Register "C."	Closed tube at rear "D."	Open hole at rear "E."	
2/ 6/18	154	240	305	610	151	370	5	30.2	74.0	2540	2333	.....	2269	2454	
2/13/18	145	212	290	525	145	313	3	48.3	104.3	2527	.....	2444	2485	2564	
3/15/18	95	120	420	670	325	550	11	29.5	55.0	2657	3074	2801	.....	.....	End of side tube (B) broken off.
4/ 1/18	112	162	500	680	388	518	10	38.8	51.8	2651	2847	2804	.....	.....	End of side tube (B) broken off.
4/ 2/18	210	280	470	630	266	350	7	38.0	50.0	2809	2885	2840	.....	.....	
4/27/18	100	128	415	588	315	460	5	63.0	92.0	2555	2989	2894	.....	.....	End of side tube (B) broken off.
5/ 7/18	135	182	490	670	355	488	8	44.4	61.0	2689	2911	2818	.....	.....	End of side tube (B) broken off.
5/ 8/18	120	135	335	440	215	305	3	71.6	101.6	2740	2748	2882	.....	.....	
6/26/18	90	92	389	587	299	495	24	16.2	23.6	2983	2591	2915	2482	.....	

Northrup Company. Fig. 2 shows the location and method of observing temperatures in four parts of the furnace. The accuracy of this pyrometer at temperatures near 3,000 degrees F., under black body conditions is estimated to be within plus or minus 20 degrees F. When observing the direct flame temperature, the variations in consecutive readings under steady conditions is within plus or minus 50 degrees F. The optical pyrometer has recently been calibrated against a standard lamp of known intensity and found to read 30 degrees F. low at 3,000 degrees. This is an error of 1 per cent which is within the probable error of temperature readings. Therefore the various temperature readings in the furnace were uncorrected.

"*Results of Tests.*—Between February 5, 1918, and June 27, 1918, inclusive, data was taken during 28 test runs on the boiler. These runs varied in length from one hour to twenty-four hours, and were made at varying rates of combustion from 3.669 to 11.276 pounds per cubic foot of furnace volume. The following different types of air registers and burners were used: Bureau Standard Natural-Forced Draft, Peabody and the Standard Forced Draft (Cramp modification) Registers. [Table 1 shows test data compiled.]

"Figs. 3 to 10, inclusive, are curves showing the rise in temperature plotted against time. These nine curves were selected, from the 28 runs made, as representative of average conditions. During the period covered by this report the furnace has been in operation a total of 302 hours. The furnace temperatures and the degree of heat saturation due to rates of combustion on these tests were more severe than will probably be obtained in service.

"Fig. 10 covers the longest period of time and the rate of combustion throughout the run was high. After about 12 hours steaming the temperatures through both the crushed brick and the Sil-O-Cel became practically constant. Near the end of the run the temperature dropped

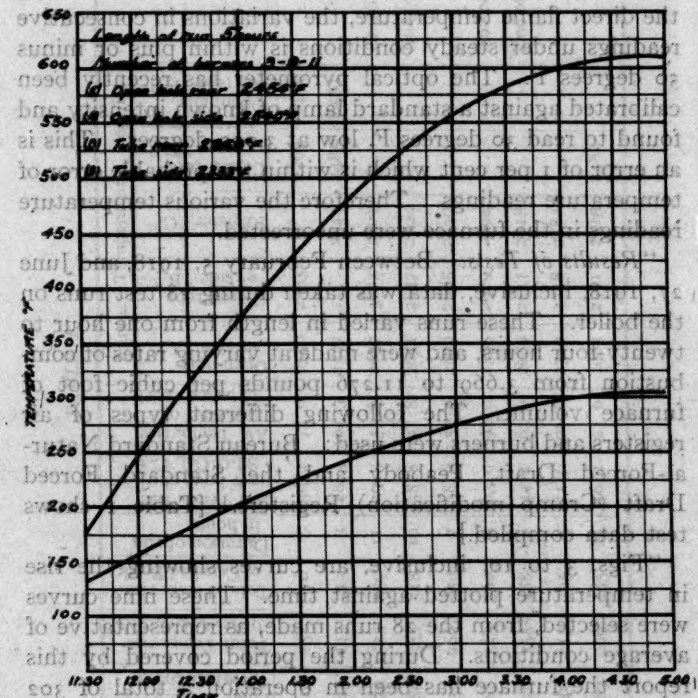

650
600
550
500
450
400
350
300
250
200
150
100
TEMPERATURE °F
11.30
12.00
12.30
1.00
1.30
2.00
2.30
3.00
3.30
4.00
4.30
5.00
Time

FIG 3

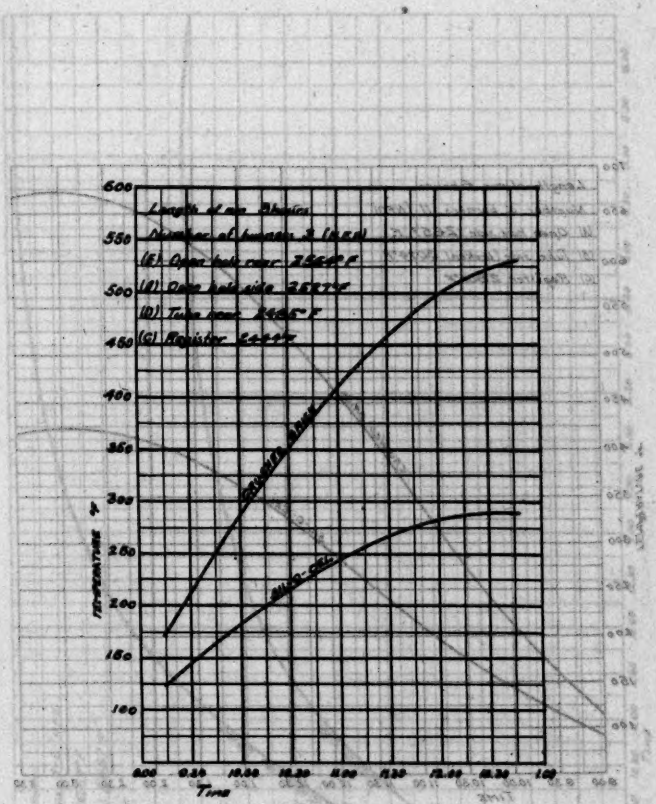

600
550
500
450
400
350
300
250
200
150
100
TEMPERATURE F
Time
Length of run 8hours
Number of burners 3 (1.2.3)
(E) Oven back rear 2660°F
(A) Oven back side 2875°F
(D) Tube front 2465°F
(C) Register 2225°F
Boiler Side
Oven Side

FIG. 4

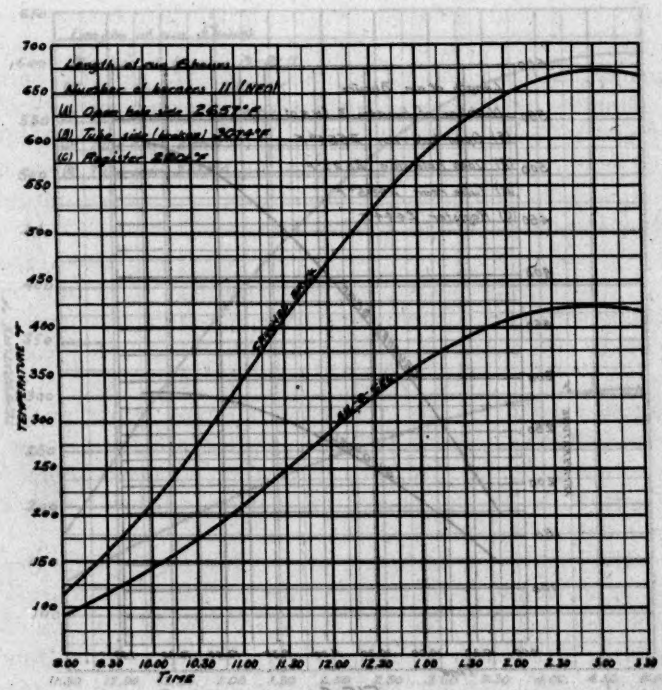

700
650
600
550
500
450
400
350
300
250
200
150
100
TEMPERATURE °F
Length of run 6 hours
Number of hearths 11 (arch)
(A) Open hole side 2657°F
(B) Take side (bottom) 3074°F
(C) Register 2804°F
GENERAL FIRE
ARCH FIRE
8.00 8.30 10.00 10.30 11.00 11.30 12.00 12.30 1.00 1.30 2.00 2.30 3.00 3.30
Time

FIG. 5

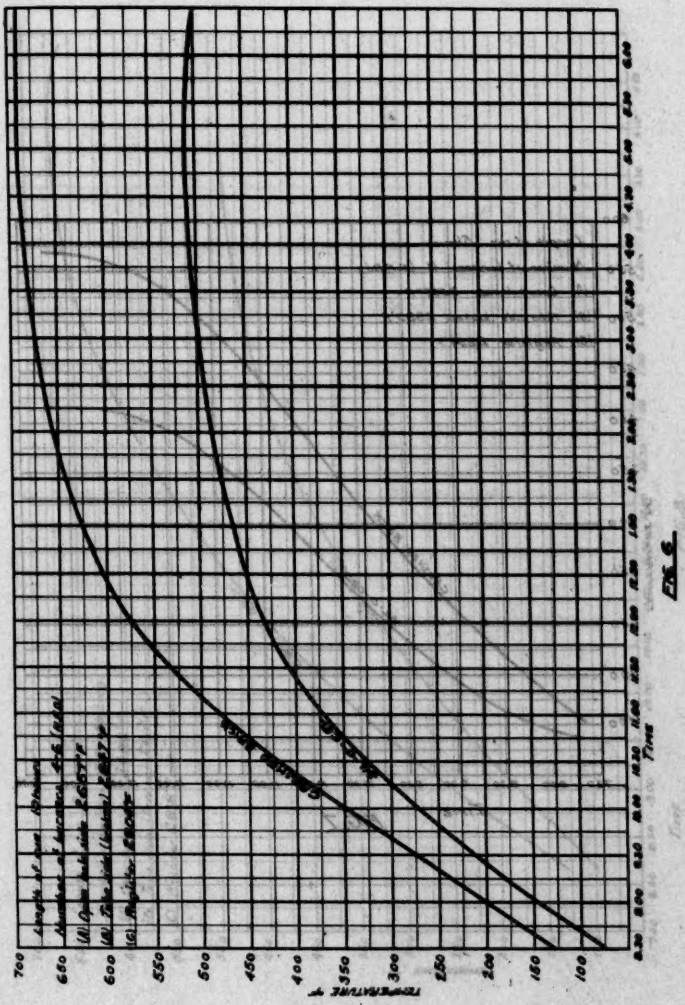

Length of run 40 feet
Number of Nozzles 4-5 (total)
(a) Pipe Insulated 2 coats
(b) Pipe not (bare) Exposed
(c) Furnace Empty
Time
Temperature °F
700
650
600
550
500
450
400
350
300
250
200
150
100

FIG. 6.

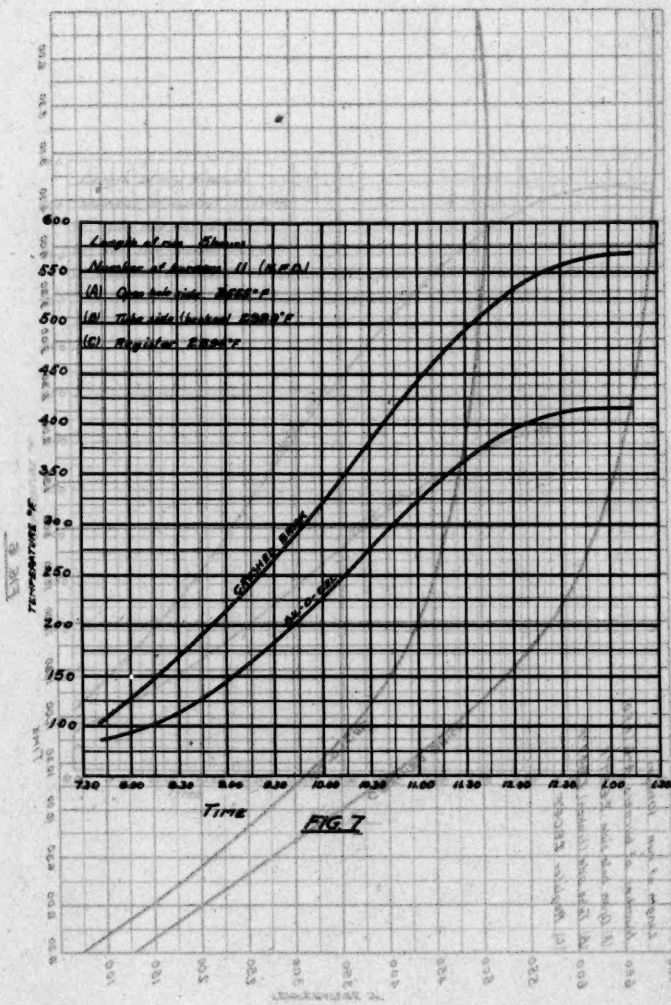

600
550
500
450
400
350
300
250
200
150
100
TEMPERATURE °F
7.30
8.00
8.30
9.00
9.30
10.00
10.30
11.00
11.30
12.00
12.30
1.00
Time

FIG. 7

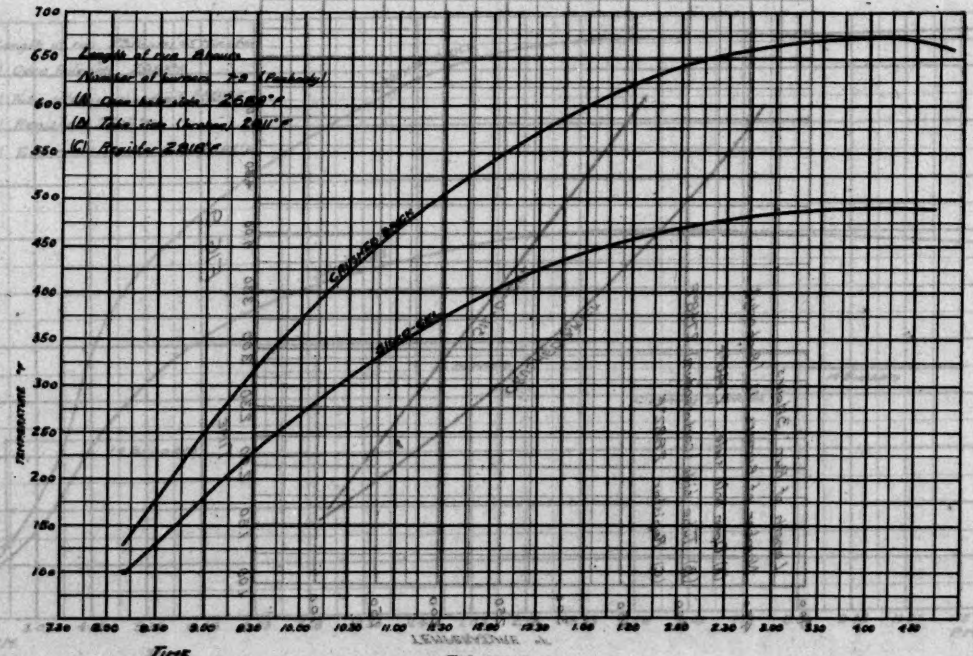

Length of run Always
Number of burners 2-8 (Packaged)
(A) Oven hole size 2'6"ø
(B) Tube size (Inches) 2½"ø
(C) Register 2'6"ø
Centre duct
Side duct
Temperature °F
Time
700
650
600
550
500
450
400
350
300
250
200
150
100
7.30
8.00
8.30
9.00
9.30
10.00
10.30
11.00
11.30
12.00
12.30
1.00
1.30
2.00
2.30
3.00
3.30
4.00
4.30

FIG 8.

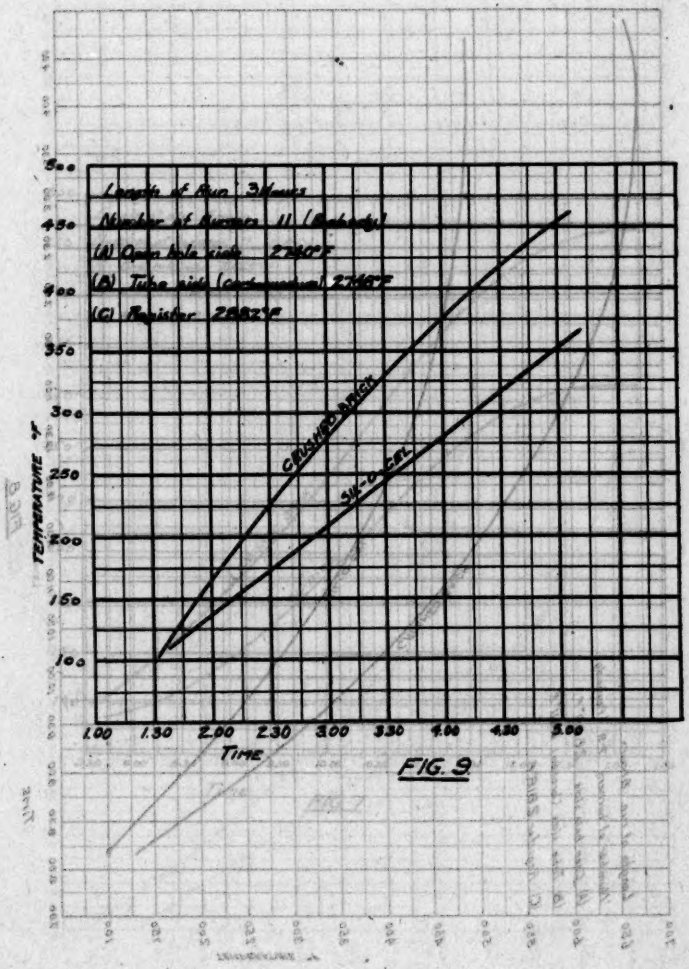

Length of Run 3 Hours
Number of Burners 11 (Each side)
(A) Open hole side 2780°F
(B) Tube side (centre row) 2748°F
(C) Register 2882°F
CRUSHED BRICK
3½-0 CRT.
TEMPERATURE °F
TIME
1.00
1.30
2.00
2.30
3.00
3.30
4.00
4.30
5.00
100
150
200
250
300
350
400
450
500

FIG. 9

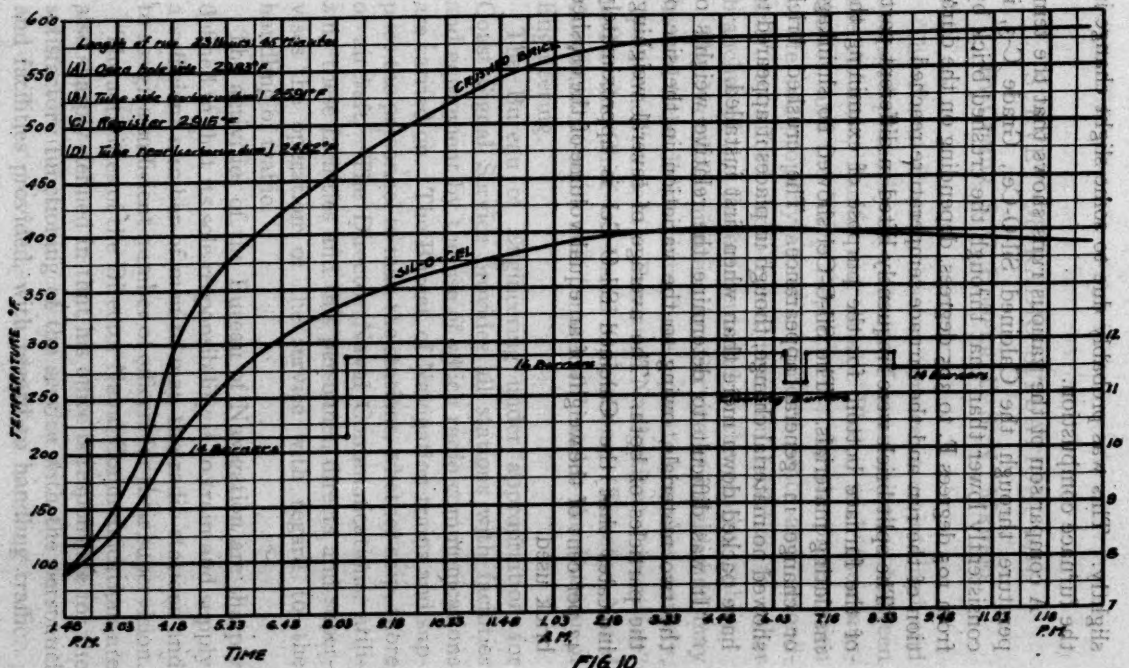

Length of run 23 Hours 45 Minutes
(A) Oven hole site 2083°F
(B) Tube side (extrapolated) 2591°F
(C) Register 2915°F
(D) Tube reproduction (extrapolated) 2462°F
CRACKED BACK
SILO-O-CEL
16 Barrels
Stoking Burners
16 Barrels
16 Barrels
TEMPERATURE °F
OIL BURNED PER CU.FT. OF FURNACE VOLUME PER HOUR (POUNDS)
TIME
1.40 P.M.
3.03
4.18
5.33
6.48
8.03
8.18
10.33
11.48
1.03 A.M.
2.18
3.33
4.48
6.43
7.18
8.33
9.48
11.03
1.18 P.M.

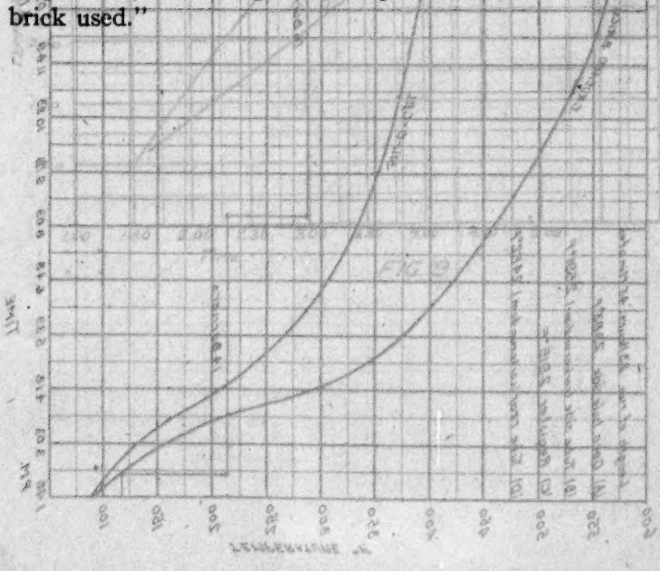
FIG. 10.

slightly. This was probably due to some slight change in the furnace combustion.

"A comparison of the various runs shows that the temperature through the Calcined Sil-O-Cel, Grade C-3, is consistently lower than that through the crushed brick by from 105 degrees F. to 305 degrees, depending on the duration of the run and the furnace temperature reached.

"The split brick were frequently lifted in different parts of the furnace bottom for the purpose of examining the insulating materials. The Sil-O-Cel showed no shrinkage or change in general appearance. The crushed brick showed no material change, though in places it appeared to have packed down more than when first installed.

"It was difficult to determine the relative weights of the two materials, owing to the variation in the size of the particles of both. The average of several weighings indicates that the Calcined Sil-O-Cel is approximately 42 per cent of the weight of an equal volume of the crushed brick used."



SURVEY OF RADIO SITUATION, 1921.

By Comdr. S. C. Hooper, U. S. N., Member.

The Naval Communication Service is maintained and operated primarily as a means of communication between the Atlantic, Pacific and Asiatic Fleets, with their auxiliaries, and the Navy Department; for the preservation of life and property at sea in time of peace and, secondarily, to afford radio communication facilities to other branches of the Government and to the general public.

The Naval Communication Service as established may be said to be divided into three separate branches, namely, "personnel" under the Bureau of Navigation; "operation" under the Chief of Naval Operations, (Director Naval Communications), and "material" under the Bureau of Engineering.

The Bureau of Engineering under appropriation for Coast Signal Service provides all stations with facilities and equipment by the use of which radio communications are carried on. The Bureau of Navigation trains and supplies the personnel to man the stations, whether ship, shore or aircraft. The Director Naval Communications, utilizing these facilities and this personnel, directs and supervises the operation of the service with regard to the handling of traffic.

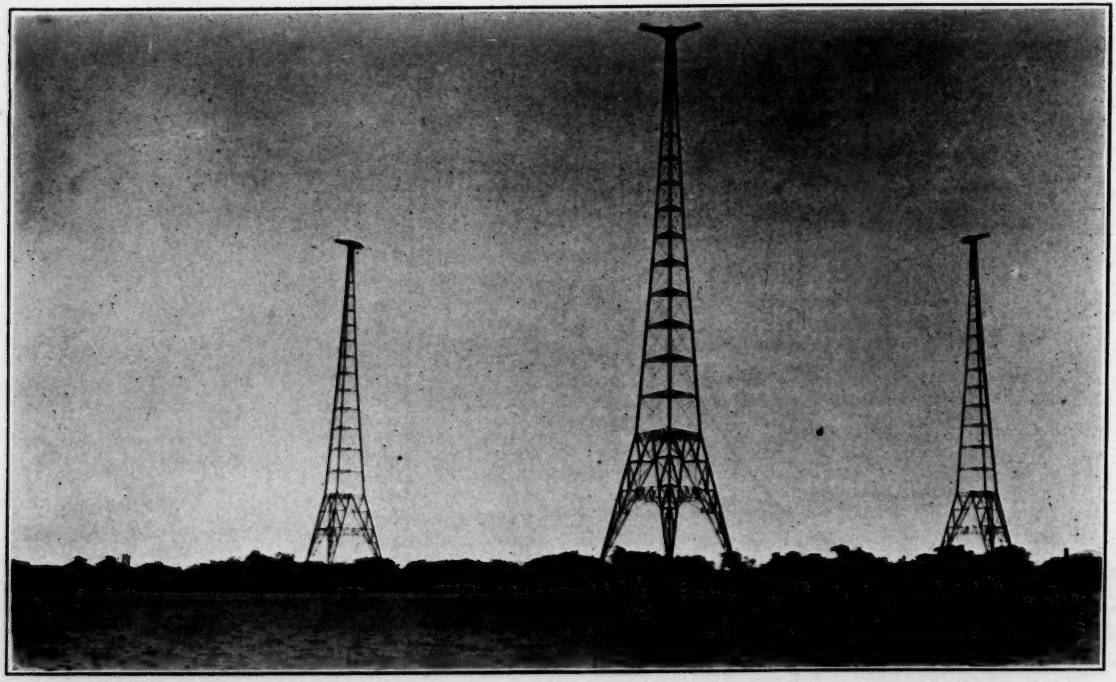
The activities of the Bureau of Navigation are sharply defined in that its sole responsibility is to train and supply a sufficient number of men to man the radio stations and to assign a sufficient number of officers for their supervision.

The activities of the Director Naval Communications are also sharply defined in that his office is responsible for the satisfactory functioning of the service with the personnel and facilities provided, with regard to handling traffic.

The activities of the Radio Division, Bureau of Engineering, are less sharply defined, and as a matter of fact its responsibilities cover a broad field of activities. These range from the conception of an idea to the accomplishing of facts of great magnitude and importance, such as the development of wireless communications from the time it was demonstrated that communications could be successfully carried on, without the use of connecting wires, between distant points beyond the range of visibility, to the time when the Atlantic and Pacific oceans and the North American continent have been spanned by radio waves of sufficient intensity to insure satisfactory and reliable wireless communications, between points of such great distances, thereby enabling the Department to keep in touch with units of the Navy in practically any part of the world.

The facilities provided by the material branch of the Naval Communication Service comprise radio stations on board practically all ships of the Navy from battle ships to tugs and the smallest auxiliary craft; chains of coastal shore radio stations along the Atlantic and Pacific Coasts, on the Gulf of Mexico, along the Great Lakes, in Alaska, in the Canal Zone, in the West Indies and in the Pacific. Also, a chain of high power trans-ocean and trans-continental stations is provided which enables continuous communications to be carried on, regardless of any other existing means of communication, such as cables, between Washington and our possessions in the Philippines, Guam, Tutuila, Hawaiian Islands, Alaska, the Canal Zone, the West Indies, and with stations in practically any country of the world which possesses stations of sufficient power to communicate with high power stations of the Naval Communication Service.

These ship, shore and aircraft radio stations are not only equipped for radio telegraph service but radio telephone and radio compass service is also either provided at



VIEW OF TYPICAL SUPER-HIGH POWER RADIO STATION, U. S. NAVAL RADIO STATION, PEARL HARBOR.

a large number of the stations or is being provided. Sound Signaling devices are being provided, the latter particularly on shipboard stations. Where the volume of traffic warrants it, the shore stations are operated duplex by means of remote control stations which enables traffic to be received at a station when it is transmitting.

The Naval Communication Service, as stated previously, is maintained and operated primarily for the use of the Navy. Its Service to provide communication facilities with our outlying possessions, independently of the cables where cable service is available, is gradually becoming more and more extensive and of greatly increased importance. This is particularly the case where the cable service is under foreign control. The Communication Service is, of course, of primary importance with regard to the safety and preservation of life and property at sea and to our ever-growing merchant marine. The Communication Service is also of the utmost importance to the general public, in that it provides facilities for the handling of commercial traffic between ships at sea and stations along the coasts situated in more or less isolated localities, in which general localities none of the commercial radio companies will maintain coastal stations due to the fact that the station receipts would not show attractive monetary returns. If the government did not maintain stations for the Fleets in isolated localities, radio communication facilities on shore would not in many cases be available for shipping.

At the present time practically all existing coastal radio stations in the vicinity of ports of New York, Boston, Philadelphia, New Orleans and San Francisco are maintained by commercial companies where the volume of radio traffic is heavy and the monetary returns fairly large. A number of high power stations are maintained and operated by the Radio Corporation of America, namely, New Brunswick, N. J., Tuckerton, N. J., Marion, Mass., Bolinas, Cal., and Kahuku, Hawaii. Other stations are to be established by this company in the United States and abroad. The

Radio Corporation's high power stations were taken over by the Naval Communication Service during the war and were either operated or closed, as were also the coastal stations of all commercial companies.

The chain of coastal stations owned by the Marconi Company, now absorbed by the Radio Corporation, were purchased by the Navy during the war as were also the coastal and high power stations of the Federal Telegraph Company. This was done with a view to eliminating, as far as possible, a multiplicity of stations within various general localities, to reduce or eliminate the very troublesome interference, and at the same time maintain a chain of shore stations which would meet all the requirements of official and commercial traffic; also to reduce the expense to the government in paying high rentals for the commercial stations taken over by the Navy for the duration of the war.

For all practical purposes, as matters stand today, there are only two radio organizations of any size in the United States capable of efficiently handling general radio traffic, namely, the Naval Communication Service and the Radio Corporation of America, the facilities of the Radio Corporation being practically limited to trans-ocean traffic. The Radio Corporation is considerd a valuable potential asset to the Naval Communication Service in that its stations, in the United States and our outlying possessions, are available for use by the Government during time of war and this Corporation is enabled to, and plans to, establish high power stations in various foreign countries which will communicate through its American stations, thereby affording the most satisfactory means of communication between American ships and American interests in foreign parts with the United States in peace times.

The Federal Telegraph Co., a 100 per cent American Company, has recently acquired valuable concessions for trans-Pacific radio service to China, and for intercity service in

China. The outlook is most favorable for this company in the Far East, and its activities promise valuable aid to American interests in that important part of the world.

The International Radio Telegraph Company has recently established a few coastal radio stations on the North Atlantic coast.

At the time the United States entered the world war the majority of the stock of the Marconi Wireless Telegraph of America was held in England and, therefore, the policy and operation of this company was dictated from abroad. The Navy Department particularly desired that a strictly American company control such of the high power radio stations in the United States as were not owned by the Government because it was found impracticable for the Navy to monopolize long distance commercial traffic, owing to the unfavorable attitude of Congress. In any event, as it would have been impracticable for the Government to erect high power radio stations in foreign countries, to operate with its high power stations in this country and its possessions, the Navy Department advocated the formation of a responsible American commercial company for high power work and eventually this policy led to the acquirement of the English interests in the Marconi Company by the General Electric Company and the formation of the Radio Corporation as successors to the old Marconi Company. It is now assured that the policy and operation of this Corporation are dictated by American citizens, with the result that it is a valuable potential asset, as stated before, to the Government, and particularly to the Navy.

The Radio Corporation has working agreements with concerns operating high power stations in England, France, Italy, Norway, Germany, Poland and other European countries and stations are to be established by this Corporation in South America. The Radio Corporation's trans-Pacific circuit communicates with Japanese stations and other stations in the Orient and further extension of

service in the Orient is contemplated. Hence there is now established a strictly American radio service with foreign countries which is rapidly being extended and developed.

The material branch of the Naval Communication Service is constantly endeavoring to improve facilities and to introduce improved methods and equipment for the radio service and every effort is being made to keep in the lead of foreign countries in this respect and thereby give our Navy every possible advantage.

Important advances have been made during the last few years all along the line. Improved transmitting and receiving equipment has been developed and placed in service, thereby increasing the effective range of ship, shore and aircraft stations. Improved devices have been developed which eliminate static and interference to a large extent under certain conditions, thereby making the service more dependable during the summer months when heavy atmospheric disturbances prevail and making the interference from interfering stations less troublesome. The Radio Compass has been developed to such an extent that this equipment has been installed in ship, shore and aircraft stations and the compass service is now depended on for ships and aircraft positions and bearings as a matter of course. A radio piloting cable system has been developed to such an extent that this system is now available for guiding vessels into harbors during fogs and at other times when navigation would otherwise be dangerous or impossible.

The Radio Piloting cable service has been turned over to the Bureau of Lighthouses for installation and operation as not coming strictly within the scope of the Naval Communication Service. The 1000-kilowatt Lafayette Radio station, established near Bordeaux, France, as a war project was placed in commission and turned over to the French Government on December 18, 1920. This station is twice as powerful as any other existing radio station and

its signals can be heard practically over the entire world, as indicated by the results of tests conducted.

The Bureau of Engineering has always advocated the placing of all ship to shore radio service, both official and commercial, in the hands of the Naval Communication Service, because, from its long and intimate experience with the radio service, it has been convinced that more satisfactory and reliable service, with less interference, can be given by one capable organization than by several, and that the Naval Communication Service is the logical service to take over this work as it represents the interests most vitally concerned, has an extensive and adequate chain of coastal stations of all types ranging from low power to super high power and must necessarily maintain this chain of stations for communication with the Fleets in any event. It is realized that for International radio service the super high power stations, particularly in foreign countries, must necessarily be established and operated by private commercial concerns and it follows that such private concerns must also have corresponding high power stations in this country.

The most satisfactory arrangement both for the efficiency of the Navy and the economy of the public would be to have Government ownership for all coastal shore radio stations handling ship to shore radio traffic and to have one or more responsible American radio concerns encouraged as much as practicable in building up, maintaining and operating a high power trans-ocean service. Ship to shore traffic can only be handled efficiently by a monopoly, either government or private. The traffic is not sufficiently lucrative to make a private company build stations except in the vicinity of the principal ports, therefore, a government monopoly is favored. As the Navy has its own stations the Navy Department becomes the logical Department to handle the traffic.

The chain of Naval low, medium, high and super-high power stations has been developed to the point where these stations can readily and satisfactorily handle all radio traffic that may be offered. Where necessary the stations have been distantly controlled for duplex operation, that is, remote control and receiving stations have been established from 3 to 40 miles from the transmitting stations with land line connections between these stations thereby enabling the operators at the remote control and receiving station to receive from one or more stations at the same time they are transmitting to one or more stations.

PERSONNEL.

At the preset moment both branches of the Service are greatly undermanned, the Operating personnel, both officer and enlisted, being 50 per cent short of the required number and the material branch being in a worse condition.

The cause for the shortage of technical men is the higher pay on the outside than in the Navy. While both branches are badly handicapped by this shortage in personnel, the Material branch is probably worse off than the Operating branch, because of the fact that it takes longer to train a good material man that it does to make a radio operator.

There are at present employed under the Bureau of Engineering throughout the service on radio material upkeep 68 officers, including Officers-in-Charge shore stations, 86 civilians under the Civil Service Commission and approximately 200 unclassified civilian radio electricians employed by the Yard Labor Boards at the various Navy Yards.

Upon this personnel rests the responsibility of keeping in condition radio and sound material valued at twenty-five millions of dollars, developing new and better apparatus, purchasing new equipments and making installations on ships, shore stations and aircraft belonging to the Navy.

Of the 68 officers referred to above, by far the greater number are temporary officers and radio gunners with a comparatively small amount of college education. For this reason the advancement in this branch of the Navy rests with the handful of Regular Navy Officers with a Naval Academy education and an extremely small number of high-class civilian employes working at pay not at all commensurate with their knowledge or duties.

In addition to the fact that at present there are not enough regular Navy officers of the proper qualifications to handle the work efficiently, there does not appear to be any source of supply of these officers save through the Post-Graduate School at Annapolis and, even in this way, in very small numbers.

The distribution of duties for material is as follows:

Bureau of Engineering.—Requisitions for new equipments, specifications and installation drawings, appropriations, patent infringement cases, shipment orders of material for installation, administrative, design, purchase and research.

Naval Radio Laboratory—Bureau of Standards.—Experimental and research work.

Naval Aircraft Radio Laboratory, Anacostia.—Experimental and research work, mainly on aircraft.

Engineering Experiment Station, Annapolis.—Sound research, experimental and development work.

U. S. S. Ohio.—Radio experimental Battleship loaned to the Bureau of Engineering for the purpose of making all sorts of practical radio experiments.

Washington Navy Yard.—Laboratory for testing and experimental work, also used for manufacture of certain radio equipment. Store-house for certain radio material. All Navy Yards have testing laboratories for radio equipment and for instruction of ship personnel.

All Navy Yards make installation of and repairs to radio and sound material on ships, distribution of material to,

and repair and installation work when necessary on shore and radio compass stations within the Naval District in which the Yard is located. In each Naval District there is a District Radio Matériel Officer who, under the Engineer Officer of the Yard, is the Bureau of Engineering's representative in that District particularly in charge of all the radio material activities. In addition to the District Radio Matériel Officer above, there is a certain complement of officers (commissioned and warrant) assigned as his assistants, in each case the complement varying with the importance and amount of work required. There is also a certain complement of civilian Civil Service personnel assigned to each Naval District who, together with the officers referred to above, make all inspections of radio apparatus on the ships and shore stations in that District, make installations and have general charge of the upkeep of all radio and sound apparatus.

The office of the District Radio Matériel Officer is always in the largest Navy Yard or Naval Station in the District, for example, the Boston Navy Yard in the First Naval District and the Norfolk Navy Yard in the fifth. A certain number of radio electricians are employed by the Yard Labor Board in each Navy Yard, and work with the District Radio Matériel Organization on repairs and installations.

From the above brief outline of the duties and organization of the Matériel Branch of the Naval Communication Service and the statements about lack of commissioned personnel it will be seen that there are a great many opportunities for officers having radio experience to work up in a large and growing organization.

Few realize the growth of the Communication Service as compared to the other branches of the Navy, such as Ordnance and Engineering. In 1912, in the Navy Department, the Division charged with the administration of operation and matériel for Communication consisted of

two officers, one civilian radio expert and one stenographer. These four occupied one corner of an ordinary sized room. Today the total space occupied by the operating and matériel branches of Communications is nearly as large as either the Bureau of Ordnance or Bureau of Engineering, and is growing rapidly, whereas these standard Bureaus have about reached their maximum strength.

Considering the fact that so much depends on Communications and its natural growth, due to the advances of science, it can be seen that the opportunities in this branch for trained officers are inviting.

The Bureau is providing the Navy, as appropriations permit, with the most modern apparatus. Each year the apparatus becomes more complicated due to additional requirements of the service. With each new class of ships, with each new naval weapon, such as aircraft for example, additional simultaneous communications must be carried on. The apparatus provided can do the work if commanding officers get together and insist on the proper execution of regulations, with properly trained personnel to handle the apparatus.

Lack of proper personnel sometimes makes us wonder whether we should not return to older and more simple systems. This, however, would be a step backward, and if we followed such a course other navies would immediately gain an advantage. The only thing to do is to keep on furnishing the most modern installations with the hope that gradual pressure from the Fleets will be the ultimate cause of solving the personnel situation. It is essential continually to strive for the ideal, and not to stop short of our highest hopes, if we wish to maintain first place in efficiency.

RADIO TRANSMITTERS.

The usual radio transmitting station consists primarily of two distinct parts, namely, the antenna and ground system, and the transmitter.

The antenna and ground system consists of the overhead part which is usually composed of a number of wires spread out to cover a larger or smaller area of ground in accordance with the rating of the station and the somewhat similar underground part known as the ground network, the two being connected by a rat-tail through the transmitter. The antenna-ground system serves actually to carry the high frequency oscillating current which radiates the power that is received at the distant station. The transmitter comprises the power system which is used to generate the high frequency currents required in radio work. The transfer to the antenna system is made through a transformer closely related in its characteristics to the usual power transformer. An important difference between these two is that the transformer for the radio set utilizes what is known as the resonant effect obtained from the inductance and capacity of the antenna to build up the antenna current. Thus we have in a radio station the power apparatus which generates the necessary high frequency current and the antenna system which transfers this energy to a circuit capable of transmitting a portion of it in the form of electro-magnetic waves that are capable of affecting receiving equipment at the distant receiving station.

The characteristics of an antenna system are very similar to those of the ordinary condenser with which we are all familiar. The overhead part constitutes one plate while the ground system constitutes the other plate and the current surges back and forth between these two plates. The value of the radio frequency current is rapidly decreased by resistance and it is therefore important to provide for a minimum resistance by making both the overhead and the underground part of a suitable number of wires so distributed that the ground system may allow a full distribution of the current along all parts under the overhead section without passing through high resistance such as rocky or

dry strata of earth. A very important feature that concerns the transmitting range is the height of an antenna and this accounts for the tendency to erect high steel towers at our modern stations. The other most important factor in determining the station's range is the radiated current. Hence it is that, as the range of a station increases, the tendency is to erect a higher antenna in order to reduce the flow of the current required or to increase the current and use an antenna of larger area, depending on the price of towers and power.

The various types of transmitters in use are classified as spark, arc, high frequency alternator, frequency changer, and tube. These may be again divided into two general classes—the spark or damped wave transmitter and the continuous wave transmitter.

The spark transmitter was the type first developed and it is still largely in use, especially for small power installations. The various continuous wave types are of comparatively recent development and are found to be superior in a number of respects to the damped wave type. Briefly, the distinction between the two is that in the spark type intermittent groups of highly damped waves (wave trains) are transmitted while with the continuous wave type the waves have an unvarying amplitude, are not grouped but are sent out continuously. The telephone radio transmitter of which we are hearing more every day is adapted to the continuous wave type of transmitter only.

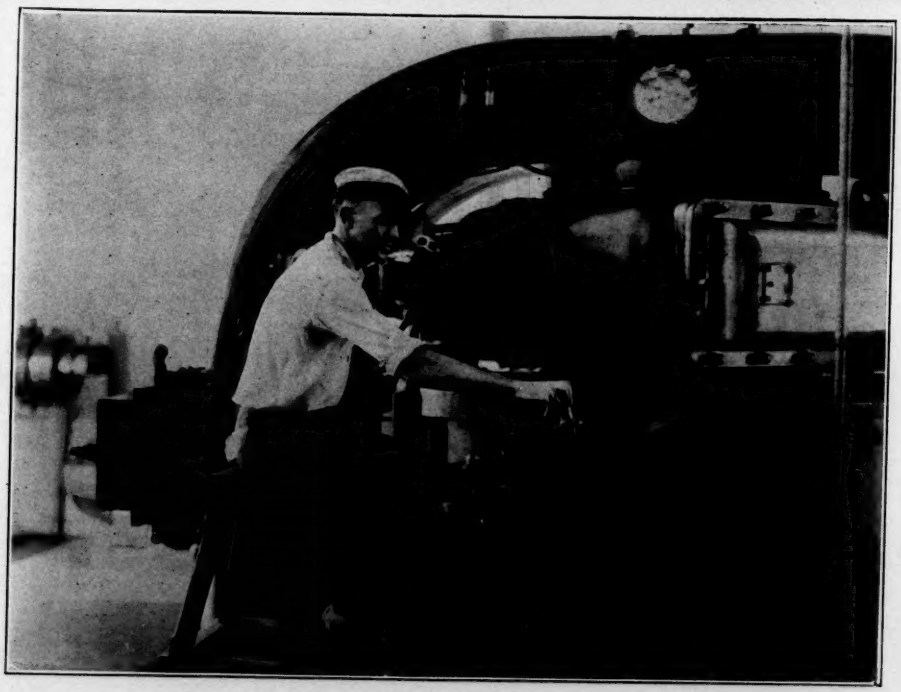
There are approximately 130 Naval shore radio traffic stations equipped with apparatus of various powers up to and including 30 kilowatts located in the territories of the United States. These stations have been gradually established over a long period and contain mostly the spark type of equipment. The oldest installations were of the early Marconi type where the power was taken directly from the commercial power supply at 60 cycles and fed into a transformer which stepped the voltage up to approxi-

mately 30,000 volts. The condensers, which were charged by a transformer to this voltage, discharged across a fixed gap or, in later installations, across a rotary gap with a great deal of noise and a very low efficiency. The high frequency current set up in this circuit, known as the closed circuit and comprising the spark gap, condensers and small inductance, was transferred to the antenna circuit and the signal transmitted had the characteristic 60 cycle low rough note.

Further investigations developed that a higher frequency was much more effective in carrying the signal and a frequency of 500 cycles was adopted as best suited for the purpose. This necessitated the introduction of a special motor generator set for the higher frequency. A further development has been the substitution of quenched spark gaps for the open and rotary types. The quench gap consists of a series of parallel discs with flat surfaces separated 1/100 inch by insulating gaskets, about 12 such gaps being commonly used in series. Its effect is to reduce the interaction between the antenna circuit and the circuit containing the spark gap and condensers after transfer of energy to the antenna circuit has been once made. The 500 cycle quench gap transmitter in sizes from ¼ to 10 kilowatts is now standard for ship and shore installations. For larger powers the arc type of transmitter is generally employed.

There are 11 stations rated as high power which are under the control and operation of the Navy. These are: Radio, Va. (Arlington), Darien, Canal Zone, San Diego, Cal., Pearl Harbor, T. H., Heeia, T. H., Cavite, P. I., Guam, Marianas, Sayville, N. Y., Cayey, P. R., Mare Island, Cal., and Annapolis, Md. These stations all have or will shortly have sets of 100-kilowatts or more. High speed sending and receiving equipments capable of 75 words per minute are being installed in all primary stations.

The first of the Naval high powered stations was that erected at Radio, Va., where a 100-kilowatts spark set was installed. This set gave very satisfactory results but failed



VIEW OF 350-K.W. ARC RADIO TRANSMITTER, INSTALLED IN THE U. S NAVAL RADIO STATION AT CAVITE.

to develop the range which was later desired. Arlington has been used for broadcasting to ships, for press and time signals and as an experimental station and served to test out the first arc transmitters, at which place the superiority of this type over the spark type was clearly demonstrated.

The arc transmitter differs radically from the spark transmitter. In the former, a direct current arc, supplied from a source developing 500 to 1000 volts, is formed between the poles of a strong electromagnet and in an atmosphere composed largely of hydrogen. The antenna is connected directly to the positive side of the arc through suitable loading coils, while the negative terminal is connected to the ground system. Power is supplied to the antenna by the arc, thereby building up a potential in the antenna system until a reverse flow of current takes place from the antenna back through the arc to the ground system. The arc then renews the antenna charge and the cycle of charge and discharge is continuously maintained, a series of uniformly spaced and equally powerful impulses being sent out.

This type of apparatus adapts itself very successfully to large power installations and has been employed exclusively in the stations erected by the Navy Department. The present installations of this type include a 100-kilowatt set at Radio, 100-kilowatt set at Darien, 200-kilowatt at San Diego, 350-kilowatt set each at Pearl Harbor and Cavite, 100-kilowatt set at Heeia, T. H., 200-kilowatt set at Cayey, 500-kilowatt set at Annapolis and 100-kilowatt sets are being installed at Guam and Mare Island. 1000-kilowatt sets in duplicate have been installed in the Lafayette station which the Navy erected near Bordeaux, France, as a war measure. The Navy has also installed a 200-kilowatt arc at the Sayville radio station, taken over from the Germans as a war measure.

Another type of transmitter developed by the General Electric Company which is suitable for high powered

installations is known as the high frequency alternator. It consists of an especially constructed alternator capable of developing the radio frequencies directly in the machine itself. A 200-kilowatt unit has been developed and very successfully used at the New Brunswick Station now controlled by the General Electric Company through the Radio Corporation of America. This machine is connected directly to the antenna and delivers a very clear signal of the continuous wave class. Signals are sent by a method of decreasing the antenna current by a device called the magnetic amplifier which reduces the radiation to a small percentage of the normal radiation. It is possible by use of this magnetic amplifier to use the high frequency alternator for radio telephony. An extremely large station based on this type of equipment is being erected by the Radio Corporation of America on Long Island for communication with Europe, South America and other points in the establishment of commercial circuits.

Two other types of equipment for the direct generation of high frequency continuous wave current are known, namely, the Goldschmidt alternator and the frequency multiplier transformer. The Goldschmidt alternator utilizes a method of building up the harmonics, always present in generators, through resonant circuits until the desired frequency is finally attained and passed on to the antenna. The transformer multiplier utilizes a special circuit which enables a given single phase current to be doubled in frequency until by two or more such multiplications the desired frequency is attained and passed on to the antenna. Such an installation is occasionally used at the Sayville Station where it was installed by the former German owners. The set is not considered very satisfactory because of the large amount of equipment and difficulties of operation. Somewhat similar criticisms as to the difficulty of operation apply to the Goldschmidt alternator which is a German development and was installed in the Tuckerton Station for some time.

The most recent development, however, is the vacuum tube transmitter. This is little more than an enlargement of the small audion tubes used in receiving work and with which we are all familiar. This system has not yet been sufficiently developed to have been installed in any of the naval stations, but the work of development is being carried forward with all possible dispatch. In comparative tests of the transmitting qualities of the spark, arc and tube type of transmitters the latter has proven its superiority. Plans are at present under way for the earliest possible installation of a set having the normal rating of 20 or 25 kilowatts at Radio, Va., for a definite and conclusive test of the operation to be expected from a large tube installation.

In the probable future Naval development of radio it is expected that tubes will directly replace the spark equipment without further considering the high frequency alternator because of the advantages in flexibility and simplicity of the tube set and the very excellent results which have so far been obtained and which are expected to be obtained in the future.

As compared with the arc transmitter the tube set offers a further advantage of very easy control and the suppression of all current in the antenna during the period the transmitting key is in the open position. With the arc sets, however, the current through the arc cannot be interrupted and it is necessary to provide some means of shunting the antenna current through an auxiliary circuit when the key is open or to let it pass continuously into the antenna and employ what is known as the compensating system of sending by which a small difference is made in the wave length of the outgoing signal when the key is in the closed and in the open position, this difference being sufficient to allow reception of the required signal at the distant station. A satisfactory system of shunting the antenna current, known as the uni-wave system, has been developed for small power arcs, but the problem of adapt-

ing the same system to the large powered arcs has not yet been satisfactorily solved although progress is being made and an early solution is looked for. Experiments to this end are being undertaken at the Annapolis Station where a 500-kilowatt set is installed and, if successful there, the system will be installed as rapidly as possible at all the other naval high powered arc stations.

The future of the high power radio station is likely to include many further developments and is already proving to be a serious competitor to the trans-oceanic cables for the transmission of commercial traffic. The radio service is already handling 10 per cent of the trans-oceanic traffic.

While great development has been made in increasing the efficiency and reliability of radio installations much work yet remains to be done. Two lines of development may be mentioned, namely, that of directive transmission which will reduce, to a very large extent, the power required for long distance transmission and means of eliminating the effect of static in receiving.

For some time the Bureau of Engineering has been working toward the modification of all spark transmitting equipments so as to provide some form of transmitter which makes use of a continuous carrier wave. Experience has shown that the distance covered, the ratio between night and day signals, and the effect of the seasons of the year on signals, is very much in favor of the continuous wave, such as those produced by arcs, tubes and machines. This type of wave also permits of very sharp tuning at the receiving station and, therefore, eliminates a good deal of interference which is ordinarily encountered when using spark transmitters.

At the present time the Navy has transmitters of the arc, spark and vacuum tube types in regular service, the latter type being divided between telegraph and telephone transmitters. In addition to the transmitters which are now in service, there is being developed a type set known as the spark tube transmitter.

Arc equipments may be divided into two classes: those designed for use on shore, and those for use aboard ship. The arcs which are designed for shore station use may be divided into two groups, Naval District or medium power arcs, ranging in power from 12 to 60 kilowatts input and High Power Transcontinental, Trans-oceanic arcs, ranging in power from 60 to 500 kilowatts. The maximum input to the arcs, however, at any of the Naval High Power Stations is approximately 350 kilowatts, the size of the antenna not permitting the use of 500 kilowatts, the maximum reliable communication range obtained with 350 kilowatts being approximately 5,000 miles. The arcs which are designed for use aboard ship may be divided into two groups, low power and high power. The low power arcs are of 2 and 5 KW capacity and are designed for use as auxiliaries to the High Power arcs or for use on ships whose antenna is too small to permit the use of greater power. These lower power arc equipments are provided with a "chopper," a device for interrupting the continuous waves so that they may be received by receiving stations equipped for the reception of spark signals. High power arcs for ships use are of 20 and 30 kilowatts capacity and are used on the larger ships where the antenna is of sufficient size. Reliable communication of 1200 miles is now possible from ship to ship and ship to shore.

The Navy has in use a large number of spark transmitters practically all of which are of the 500 cycle type and vary in power from one-half to 100 kilowatts. Each size serves a particular purpose, the smaller ones being used as auxiliary for the larger power equipments or for installation on smaller ships. The high power spark equipments are used on the larger ships and at shore stations, the 10 kilowatt size having an approximate range of 500 miles, the range being limited by the antenna height.

The spark tube transmitter which is being developed by the Bureau is a vacuum tube equipment which makes use

of a part of the spark apparatus. The filaments of the vacuum tubes are lighted from the local supply, the tube plates being supplied from the high potential side of the 500 cycle transformer. The wave emitted is essentially continuous in character but is modulated by the 500 cycle supply so that it may be received with a pure 500 cycle note on the ordinary spark receiving equipments.

Perhaps the most interesting of the Navy transmitting equipments are the vacuum tube telephone and telegraph transmitters. These equipments are built in sizes from five watts up to 750 watts capacity and are designed for the reliable transmission of the voice over distances varying from a few miles up to approximately 400 miles. The low power equipments are designed for two way conversation just the same as for a conversation over the ordinary telephone line.

The largest radio telephone equipment, rated at 750 watts capacity, is designed for use at aircraft bases. These equipments may be controlled over many miles of ordinary telephone line. That is, a conversation may be held between a plane say 100 miles from its base and a subscriber's telephone say 500 miles from the aircraft base, the voice being carried from the subscriber's telephone over the ordinary land line to the aircraft base and thence to the aircraft by radio.

The shift from spark sets to continuous wave equipment must necessarily be gradual and over a number of years as it is out of the question to replace existing equipment with new expensive equipment until the former has worn out.

It is the policy to design all receiving equipment for maximum selectivity, it being advisable to sacrifice signal strength for selectivity in the receiver and then amplify the remaining signal to comfortable receiving strength. Special circuits and equipment are used both afloat and ashore to cut out interference caused by atmospheric disturbances and disturbing transmitting stations. It may

be said that, in general, the first class receiving equipment afloat consists of a special tuning device, a receiver and a multi-stage amplifier. The shore station long distance receiving equipment is a little more complicated in that it is possible to make use of ground wires, loop and antenna combinations for the elimination of atmospheric disturbances, which were very serious when large high mast receiving antennas were in vogue.

At the present time the Navy is using three types of amplifiers which are commonly known as two-step audio, three-step audio and combinations of radio and audio. In the first and second types, that is, audio frequency amplifiers, the signal is amplified at audio frequencies. In the third type, the incoming radio frequency signal is amplified by three steps of radio frequency amplification and made audible by a third tube (detector tube) and then amplified by two stages of audio frequency amplification. The total amplification of a combination amplifier results in a signal many thousand times the strength of the original signal.

SOUND AND RADIO RESEARCH.

The Research Section of the Radio Division has cognizance over Research work on sound and radio apparatus. Whenever apparatus is submitted for consideration by the Bureau of Engineering, with reference to using this equipment in the Naval Service, details of this equipment are reviewed by the Research Section, whose duty it is to determine the theoretical and practical possibilities of this apparatus. In addition to this, information on new inventions and ideas which may be presented for consideration are reviewed by the Research Section and, whenever possible, they are subjected to careful consideration, upon conclusion of which if there is a favorable outlook for the data submitted, steps are taken whereby the various Laboratories under the cognizance of the Radio Division are

assigned problems to determine the practicability of the suggestions offered.

The Research Section of the Radio Division has under its jurisdiction three major Laboratories which are qualified to handle Research Problems and to conduct extensive investigations on these problems. These Laboratories consist of the Naval Radio Research Laboratory located at Bureau of Standards, Washington, D. C., U. S. Naval Radio Aircraft Laboratory, Anacostia, D. C., and Engineering Experiment Station, Annapolis, Md., small testing Radio Laboratories at the Navy Yards, and the new Radio Test Shop at the Navy Yard, Washington. The Radio Research and Aircraft Laboratories handle the major part of the radio research work while the Engineering Experiment Station, Annapolis, handles all the research work on sound apparatus. The Yard Laboratories handle service tests of apparatus that has been previously determined as suitable for Naval needs and preparatory to its installation on shipboard, and also are assigned a certain amount of experimental apparatus for instruction of personnel.

The general procedure pursued in handling research problems is first, upon receipt of necessary authority, to assign a problem number to the investigation and authorize any one of the above mentioned Laboratories to conduct this investigation. Problems are arranged in three classes with reference to the priority of these problems and are given letter numbers, such as "A", being the most important, "B," next in importance, and "C" next, and so on.

Upon receipt of necessary authority to conduct research work on problems assigned by the section, work is started on these problems; upon completion of this investigation, a "soap-box" model of the finished product is made. Upon notification by the Research Laboratory that the investigation is completed the Research Section informs the Design Section of this fact and submits the "soap-box" model of the equipment. The Design Section then submits the

"soap-box" model to the Washington Yard where it is remodeled to comply with service requirements. Upon completion of this work the equipment is ready to be submitted to the various manufacturing agencies who produce the equipment in quantities.

Approximately 105 radio research problems have been under investigation by this Bureau, approximately 50 per cent of which have been completed. Among the completed problems which are of an important nature are the Radio Control of Coast Battleship No. 4, (ex *Iowa*); the Radio Piloting Cable; the Radio Controlled Torpedo; Eaton Uni-wave Key for Elimination Arc Compensation Wave; Anti-Static Receiving Systems; the Improvement of Shore Radio Compass Equipment; Radio Relay by Re-radiation of Received Signal; Spark Tube Transmitting Equipment; Underwater Receiving Equipment for Submarines; Depth Sounding with Sound Listening Devices; Improvements in Sound Receiving Apparatus for Destroyers and Submarines, and numerous others of lesser importance.

The U. S. S. *Ohio* has been obtained for use as a radio experimental vessel. This vessel is equipped with the latest type radio apparatus and its duties consist of conducting various problems affecting Research, Design and Installation and Operation of this equipment on board Naval vessels. The facilities obtained on board the U. S. S. *Ohio* are such that the Bureau is constantly in touch with any desired improvements necessary to equip the apparatus to meet actual service conditions. This vessel is ordinarily stationed at Annapolis, Md., and is available for any tests of a radio nature that may require its services whether lying at anchor or cruising up and down the Atlantic Coast. The acceptance of radio apparatus after having been subjected to service tests conducted on board the *Ohio* is such that this equipment is ready for installation on board any ship in the Navy and the benefit derived from

these tests are such that the Bureau is thoroughly conversant with what may be expected of this apparatus when placed on board Naval ships.

SCOPE OF THE RADIO COMPASS.

That the system of Shore Radio Compass Stations operated by the Navy Department as an aid to navigation is proving of great value to our shipping is testified to by the hundreds of letters received by the Department from the Masters of merchant vessels. Shore Radio Compasses or Radio Goniometers, as they are sometimes known, consist of a special type of radio receiving apparatus coupled with a revolving coil and are capable of determining the direction of signals emanating from a ship's radio transmitter. These stations have been erected at carefully selected locations along the coast and in the vicinity of the principal harbor entrances and maintain a continuous watch 24 hours of the day, every day in the year.

When a ship at sea desires its bearings from one or more of these radio compass stations, she calls the station or stations in the regular way and makes a prearranged radio signal for one minute. The operator at the compass station takes the ship's bearing while she is transmitting and forwards the bearing by radio to the ship. A bearing may be had simultaneously from any number of these stations within range and by the use of two or more bearings the Master can accurately fix his position.

POLICY ABOUT LOCATING STATIONS.

The location of the individual station depends upon several important factors:

(1) The possible military importance.

(2) Advantage from the standpoint as an aid to Navigation.

(3) Surrounding characteristics that will enable satisfactory radio compass operation.

(4) Means of communication with vessels at sea, and

(5) Possibility of co-ordination with an adjacent station.

1. In the first place, it is necessary to establish a radio compass station at such a point where it will give the maximum service from a military and naval standpoint in case of hostilities. A review of the list of established stations will show that the large ports on the Atlantic and Pacific Coasts are amply provided for. In addition, certain other points of importance have been included which, when considered in the entire organization, provide facilities that will be of utmost value during hostilities.

2. In order to maintain the efficiency of this important service in time of peace, the facilities of these stations are offered to the ships of the world as an aid to navigation; therefore, consideration of this fact is given in selecting the sites.

3. There are many cases where an ideal location from a standpoint of navigation and military importance cannot be used due to the surrounding characteristics which would prevent the radio compass from functioning in the proper manner. A site must be selected which is free from surrounding metallic structures and should be, if possible, a low sandy beach, a considerable distance from any hills and elevations. Rocky terraine, elevations and metallic structures have a tendency to cause refraction or bending of the wave front which causes an undesirable deviation in the observations.

4. An extremely important point in the establishment of a radio compass station is, to locate it so as to use an existing radio transmitting station, if that be possible. Otherwise, it will be necessary to install a radio telegraph transmitter, which is an additional expense and, in addition, increases interference.

5. It is always advisable where possible, to arrange for the coordination of two or more stations in the neighborhood of a harbor entrance. In locating two or more radio

compass stations which are to be co-ordinated, the length of the base line between these two stations is a primary consideration. From experience, it has been ascertained that a distance of thirty to forty miles provides an ideal base line.

These are the important essentials to be considered in the establishment of radio compass stations and it is obvious that a mass of detailed investigation will be covered before suitable sites can be decided upon.

CALIBRATION.

When stations have been established and construction work and installation of apparatus completed, it is necessary to calibrate to correct any deviations that may exist. The method of calibration required is as follows:

A transit is set up on the compass house and the true meridian is ascertained. A vessel then maneuvers on an arc of a circle about the station at a distance of from two to five miles, constantly making radio signals which are observed by the radio compass operator. While the Radio Compass operator is making an observation, simultaneously the transit observer obtains the true bearing of the ship. This procedure is continued through the entire working sector of the station. The data compiled is then plotted and a deviation curve drawn. All the work of calibration is conducted with great care using extreme accuracy. The observations of the radio compass and transit are read to fractions of degrees, thereby producing the wonderful accuracy for which the U. S. Radio Compass Stations have been credited.

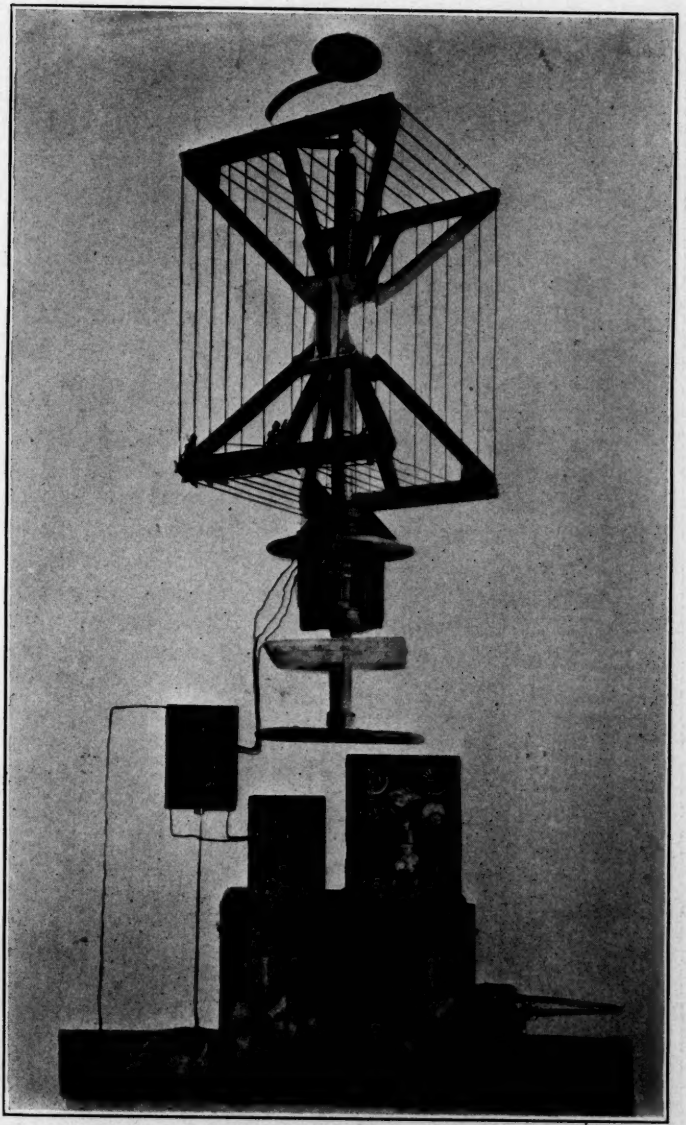
RANGE REQUIREMENTS.

The range of shore radio compass stations depends on the following:

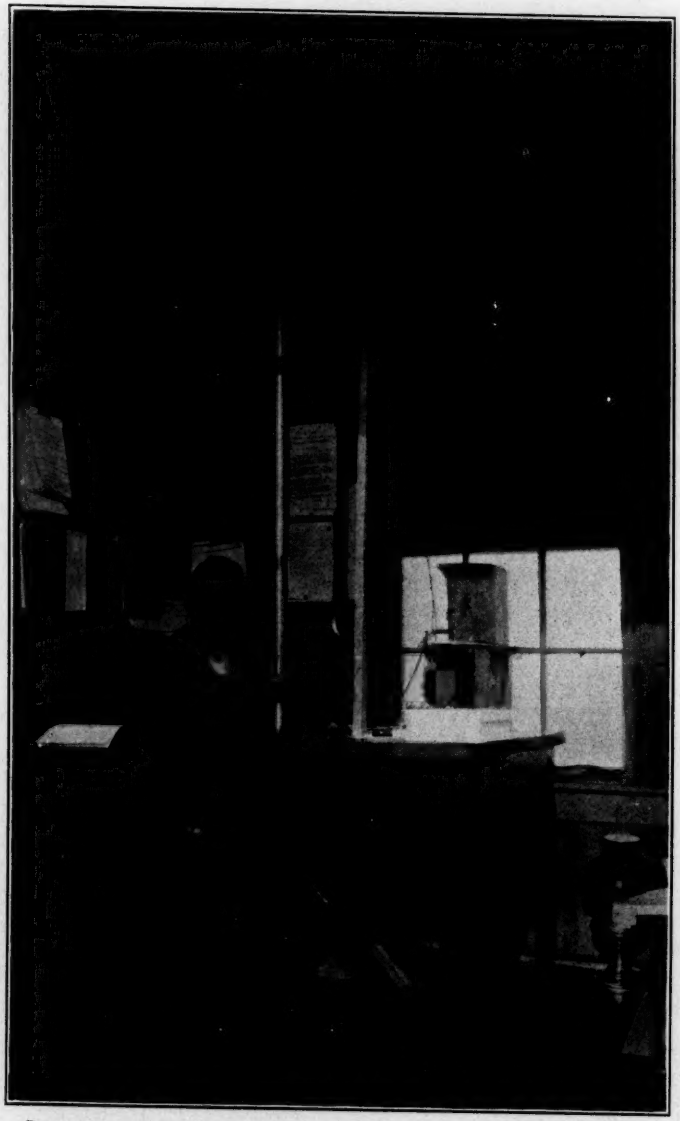
A. Sensitivity of the Radio Compass receiving apparatus and ship's receiving apparatus.

B. Power of shore and ship transmitters.

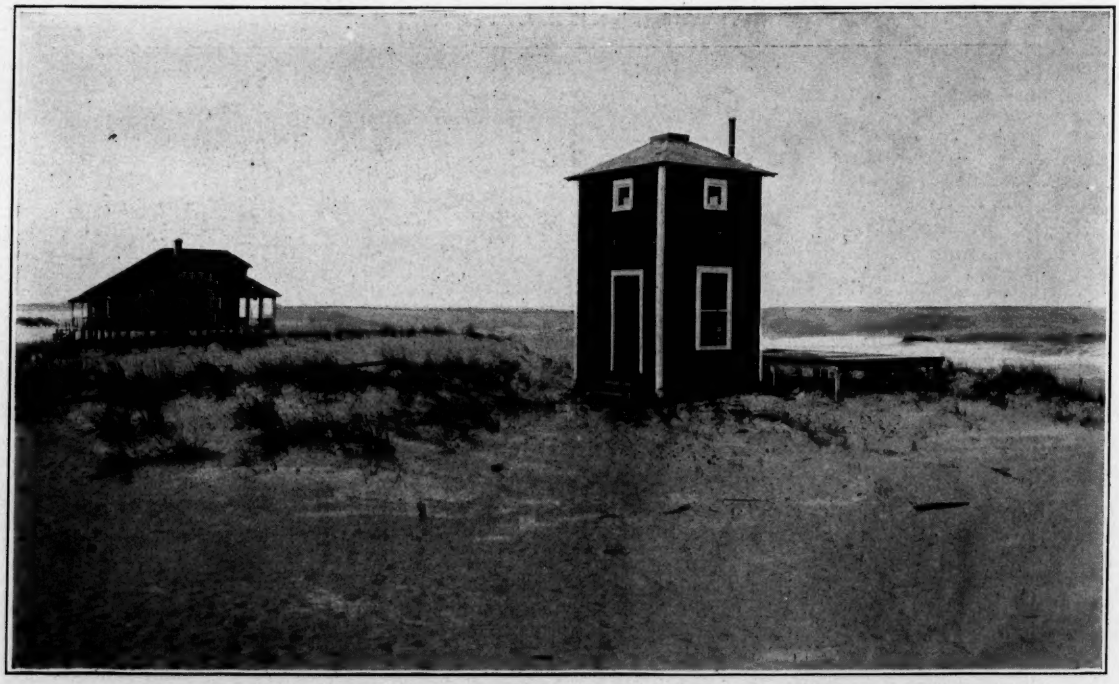
C. Accuracy of calibration.



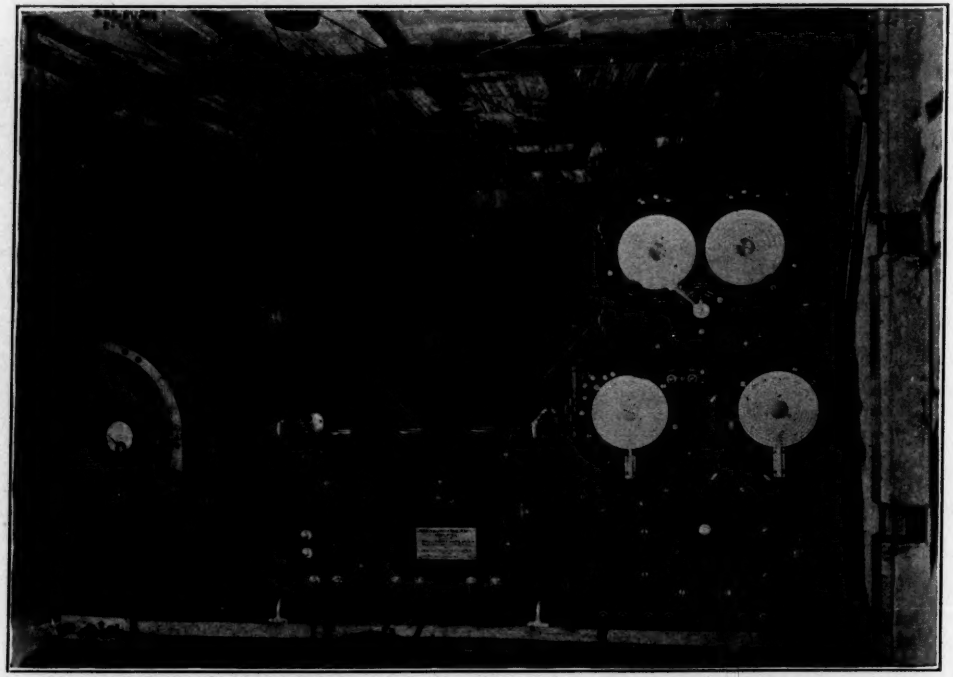
View of Radio Compass Coil and Radio Receiving Instruments.



INTERIOR OF RADIO COMPASS STATION SHOWING OPERATOR ON WATCH.



VIEW OF TYPICAL RADIO COMPASS STATION LOCATED ON THE BEACH. U. S. NAVAL RADIO COMPASS STATION, BETHANY BEACH, DEL.



RADIO INSTALLATION IN F-5 SEAPLANE.

For navigational purposes the range is limited to about 100 miles due to the fact that the majority of ship's transmitters cannot produce signals of sufficient strength for good compass work at a greater range.

However, for military purposes where ships are using high powered transmitting apparatus, ranges of 1000 miles or more may be obtained. Bearings taken at this distance, however, must be treated as great circles to avoid errors. On January 4, 1920, the American Steamship *Eastern Breeze* received a set of bearings from the New York Harbor entrance stations, which fixed the ship's position 830 miles southeast of New York. It was later ascertained from the Captain of this vessel that the position given was within one and one-half miles of its actual position.

CO-ORDINATE STATIONS.

The harbor entrance stations are operated in groups of three or more and centrally controlled from a plotting station connected with each compass station by land wire telegraph. In this case, taking for an example New York, a ship approaching New York, and desiring her position would call "NAH," the central control station located in New York City. The control station would notify by land line the compass stations at Montauk, Fire Island, Sandy Hook and Mantoloking, that a ship (giving call letters) is requesting bearing. When all stations have acknowledged, the control station, by distant control using the New York Navy Yard transmitter, directs the ship to test. The ship then transmits her call letters repeatedly for one minute, during which time the compass stations obtain bearings. The bearings observed are then given the control station by land line and the control operator roughly plots them to ascertain the accuracy of fix that has been obtained. The fix usually being satisfactory, the bearings are then furnished the ship in message form by radio.

In case of an independent Radio Compass Station the procedure for the ship is identical; but on shore the Compass Station uses a distant controlled radio telegraph transmitter for communication as has been outlined above for the control station.

NUMBER OF STATIONS.

It is an interesting fact to note that the first permanent Naval Radio Compass Stations were established in the fall of 1918, and in approximately two years the organization has expanded to the following:

23 Stations in commission on the Atlantic and Gulf,
6 Stations in commission on the Pacific,
8 Stations building and being improved on the Atlantic,
14 Stations building and being improved on the Pacific,
6 Stations building on the Great Lakes,
3 Stations additional authorized on the Atlantic,
1 Station additional authorized on the Pacific,
2 Stations building in Alaska,
3 Stations authorized for outlying possessions.

66 Total.

RECAPITULATION.

23 Stations on the Pacific and Alaska Coasts,
34 Stations on the Atlantic and Gulf,
6 Stations on the Great Lakes,
3 Stations in outlying possessions.

66 Total.

Included in the above are several stations established at Naval Air Stations, Pensacola, Rockaway, Cape May, Lakehurst, Canal Zone and Hawaiian Territory.

NUMBER OF BEARINGS FURNISHED.

As an indication of the use vessels have been making of the stations in commission, the December report, which is an average for this period of the year, will be of interest.

2,702 ships were furnished 4,250 bearings by 23 stations on the Atlantic and Gulf Coasts.

The average time required to furnish the bearings after the ship had been told to test was 4 minutes and fifteen seconds.

The Cape Hatteras Station alone furnished 497 bearings, which indicates that this station was greatly needed by mariners.

EFFICIENCY.

The efficiency of a device of this character is easily proven by the service it renders. From the December report just mentioned it is obvious that the navigators are obtaining satisfactory results from the bearings furnished.

Speaking generally, the efficiency depends first, upon the apparatus which is constantly being improved and today is equal to the Radio Goniometers used in France and the Radio Direction finders (Bellini-Tosi) used in England; second, the method of calibration which has been so developed, that during the calibration, readings are recorded to 1/10 of 1 degree; third, much depends upon the ability of the operating personnel and, although the apparatus is quite simple, efficient operation can only be accomplished by much practice and experience.

In connection with the establishment of the radio compass service the question of policy as to whether all ships should be equipped with radio compass apparatus and plot their own bearings or positions from observations made on the signals emitted by fixed stations on shore or whether sufficient shore stations should be established and plot the bearings or positions of ships from the observation of signals emitted by the ships, was decided in favor of the latter,

owing to the expense and difficulties involved in insuring installations on shipboard, particularly as regards merchant ships, and the necessity of having specially trained personnel for this service which could be more readily provided for on shore than with the changing and inexperienced operators on board ship. The results obtained justify the decision made as the radio compass service, notwithstanding the comparatively brief period since its inception, has been highly successful and is now considered as a highly valuable aid to navigation, to the Navy and the Merchant Marine and also to foreign vessels.

AIRCRAFT RADIO.

The success of aviation from a military and naval standpoint depends to a large extent on the efficiency of its communication. Today radio is the only satisfactory system of communication which the aviation service has and to a large extent the expression "aircraft for radio" or "aircraft for communication" is more nearly correct than "radio for aircraft." Aircraft radio is confronted with certain problems which do not enter into ship or shore radio communication at all, the greatest of these is the necessity of reducing weight to an absolute minimum.

On the turret spotting planes of the Fleet, the maximum weight allowance for radio equipment is 110 pounds. With this weight limitation, however, it has been possible to construct a small telephone and telegraph combination transmitting and receiving set capable of two-way communication over a distance of ten miles. A heavier and more powerful spark telegraph set, including receiver and radio compass, is installed in the large Fleet sea-planes and with it two-way communication has been attained a distance of four hundred miles, but under average conditions 50 miles is the range that can be depended upon.

A shore radio set is about to be installed at the air Stations for its primary object—communication with air-

craft—and which it is expected will communicate 500 miles telegraph and 250 miles telephone. This set will be useful for both commercial and Naval aircraft.

One of the most important of the more recent developments of aircraft radio has been the acquisition of the aircraft radio compass. With the present radio compass as now installed in practically all of the larger type seaplanes, a plane has flown out to sea taking radio compass bearings of a battleship underway at an unknown position 100 miles distant, has picked up and flown over the battleship and then turned around and flown back to its original station by means of radio compass bearings only. This proves conclusively the value of the radio compass for aircraft especially when long flights are made. The distance above mentioned will undoubtedly be greatly increased when the pilots and radio operators get more familiar with its use and operation and when, by experience, they get to place more reliance in it.

In order to get any results at all from the radio compass it was necessary first to overcome a difficulty, which a year or two ago seemed insurmountable; namely, the elimination of ignition disturbances in the receiver attached to the radio compass, caused by the presence of high tension leads running from the pilot's seat back to the motors.

Other important problems in connection with the improvement of aircraft radio communication on which a great deal of investigation and development work has been done are: Improvement of Inter-communication system; Development of a system of emergency transmission when plane lands on water; Elimination of storage batteries as a source of power for transmission with the consequent weight reduction; Amplification of received signals. Other problems with which investigation is still in the preliminary stage are: Use of the radio compass on all-metal planes; Use of radio compass on rigid dirigibles; Use of radio to permit of landing during night or in a fog.

The Aviation branch of the Navy is closely connected with the other branches of the Navy through the medium of the radio communication service. Several of the Air Stations are situated either directly at or close to the radio stations of the Primary system of the Naval Communication Service. Where a radio station of the primary or secondary system of the Communication service was not situated close to the site selected for the Air Station a radio station was constructed at the Air Station.

All radio stations of the secondary system in the Navy are equipped to communicate with aircraft and one radio station of the secondary system which is located within the Air Station grounds has regular periods during the day when it does nothing but communicate with aircraft. On the occasion of a flight along the Coast of a large sea-plane all radio stations which would be within the communication range of the plane on its trip are told to keep watch on the aircraft wave lengths.

All Navy radio compass stations along the Coast are, as in the case of the radio stations above, informed of the date of the plane's cruise along the coast in order to be able to take bearings and send the plane's position, if requested.

NOTES.

THE DESIRABILITY OF STANDARDIZATION IN THE TESTING OF WELDS.*

By F. M. Farmer, of the American Bureau of Welding, New York.

Welding by gas and electrical means has long since passed the stage of small applications, and is becoming one of the more important of the mechanic arts employed in the manufacturing and fabrication of metallic structures. Although the welding art had been developing very rapidly, expansion was particularly rapid during the war because of the great possibilities for saving time and labor which were made evident. Since the war the field of application of welding has continued to expand, being accelerated no doubt by the ever-increasing economy over other methods of joining metals, which involve any considerable amount of labor.

No argument is needed for standardization as a general policy. Its value in most fields of human activity, and particularly in engineering, is generally acknowledged. Therefore, when a new art becomes an important element in industry, standardization is not only desirable, but is almost essential if development along sound engineering lines is to continue. That this view is recognized, is indicated by the fact that one of the important functions of the Bureau of Welding—an organization established by the American Welding Society in co-operation with representatives of Government Departments and Technical Societies—is to standardize various phases of gas and electric-welding, such, for example, as methods of welding, the training of welders, tests of welds, nomenclature, methods of inspecting and testing welds in completed structures, etc.

So far as the author is aware, no welding standards of any kind have yet been proposed in the United States, either by the Bureau of Welding or other authority. However, there has been considerable discussion of this phase of the welding situation, particularly by the Welding Research Committee of the Emergency Fleet Corporation, and as wide discussion is essential to progress in real standardization, this paper is presented as a contribution to the subject of standardization in connection with the testing of welds. Standards for evaluating welds are suggested, and certain standard procedures are proposed for those mechanical tests which the author considers to be the more important. It is believed that a free discussion of the proposals from the standpoint of British practice will have a great influence in promoting standardization.

Need for Standardization in Testing Welds.—Standardization of the procedure in making a test of any kind is obviously necessary before results obtained by different observers can be compared. Tests of welds are no exception to this general rule. Differences in details of procedure have caused such widely divergent results that comparisons are frequently impossible, with the result that the usefulness of much of the research work on record is greatly restricted.

* Paper read before the Institution of Mechanical Engineers, on Friday, February 18, 1921.

The great need, however, in welding is for some standardized method of measuring or determining the relative value of a weld. The ideal weld is presumably one which does not change the characteristics of a structure containing such a joint from those which the structure would have, if no joint had been necessary. In other words, the joint should have the same characteristics as the adjoining metal. If this view be correct, what tests should be made to determine these characteristics, and how is the degree of approximation of a weld to the ideal weld to be determined?

STANDARDS FOR TESTING WELDS.

Fundamental Basis.—A weld is, in general, made for the purpose of joining together two pieces of metal in a structure. Joints are usually a necessary evil. They are avoided when it is possible to fabricate structures without them. The effectiveness of a joint is therefore determined by a comparison of the characteristics of that part of a structure containing the joint with a similar part which has no joint. Sometime in the future, processes may be developed which will produce welds having inherent characteristics, such that welds may be introduced into a structure for the sake of acquiring these characteristics. In that case the above basis of evaluation would not be applicable, but that time has not yet arrived. Also the argument has been advanced that because a weld is non-homogeneous, it cannot properly be compared with a homogeneous metal. While this is true as far as some characteristics are concerned, the fact remains that in the present stage of the art, welds are simply joints, and the practical basis of evaluation of a weld is in terms of the metal of the parts joined together.

Kinds of Standards.—It is obvious that for many purposes an investigation of a weld need not be extensive. There should be, therefore, several standards which will serve the more usual purposes. Three standards are suggested—shop standard, commercial standard, and research standard.

(*a*) *Shop Standard for Testing Welds.*—In the shop a standard test is required for such purposes as checking the work of a welder from time to time, testing a new lot of electrodes in electric welding or of welding wire in gas welding, testing the effect of some change in conditions which may have taken place, etc. Such a standard must necessarily be as simple as possible and preferably a single test. The technique of the procedure in making the test must be of a simple character. For such a standard, a bending test is suggested. A bending test involves more of the important properties of a material than any other single static test, yields the most information, and can be made with very simple apparatus.

(*b*) *Commercial Standard for Testing Welds.*—There are many cases where more than one kind of test should be made, but where the circumstances do not justify a complete investigation. There is, therefore, a need for what might be termed a commercial standard for making, for example, comparisons between different commercial welding processes or between different kinds of electrodes in electric metal arc welding or different wires in gas welding. It is suggested that such a standard consists of three tests—a bending test, a tensile test and a fatigue test. These three tests probably give more information than any other three tests that could be selected.

(*c*) *Research Standard for Testing Welds.*—When a complete investigation of a weld is to be made for research or other purposes, all tests and examinations are made which will contribute any information in regard to the characteristics of the weld. It may be necessary to make

tests of a special character in special kinds of welds or where application under unusual conditions is contemplated, but there should be a standard list of tests which will give information which is complete, as far as all ordinary requirements are concerned. The following is suggested for such a standard list:—

A. *Tests of Base or Parent Metal:*—

(1) Chemical analysis. To include carbon, manganese, silica, phosphorus and sulphur.

(2) Mechanical tests:—

(*a*) Tensile. To include yield-point, ultimate strength, total elongation, and reduction of area.

(*b*) Bending.

(*c*) Fatigue.

(3) Metallographic. Photomicrographs at 100 diameters.

B. *Tests of Weld:*—

(1) Mechanical tests:—

(*a*) Tensile. To include yield-point, ultimate strength total elongation, and reduction of area.

(*b*) Bending.

(*c*) Fatigue.

(*d*) Impact.

(2) Metallographic. Photomicrographs at 100 diameters at (*a*) the junction between the base metal and the filled-in metal, (*b*) the base metal adjacent to the junction, (*c*) the center of the filled-in metal.

C. *Tests of Filled-in Metal:*—

(1) Chemical analysis. To include carbon, manganese, silica, phosphorus and sulphur.

(2) Mechanical tests. Standard round specimens to be prepared from ingots deposited in a base-metal mould of such dimensions that normal cooling conditions will be approximated:—

(*a*) Tensile. To include yield-point, ultimate strength, total elongation and reduction of area.

(*b*) Bending.

(*c*) Fatigue.

(3) Metallographic. Photomicrographs at 100 diameters.

Numerical Evaluation of Welds.—The evaluation of articles manufactured in quantity is usually determined by comparison with a standard of some kind. Where a number of variable properties or characteristics are involved, the most reliable evaluation is obtained by employing a numerical method. For example, each property is compared with the corresponding property of the standard and a percentage value assigned. Where such comparison cannot be made by actual measurement, a value is assigned which is based on judgment. If all of the properties are of equal importance, the final evaluation figure is simply the average of the various percentages. In general, however, certain properties are more important than others, and should therefore be weighted before averaging.

In the case of welds, most of the characteristics are determined by test and expressed in a numerical result. The standard is usually the base metal, so that an evaluation of each characteristic of the weld in per cent of that of the standard is easily made. The principal difficulty is in selecting the weights to be used when averaging for the purpose of assigning a value for the weld as a whole.

In the "shop standard" proposed, only one test is made so that evolution is obtained directly. In the "commercial standard" three tests are made, the relative value of the results of which will depend upon the purpose of the investigation. If, for example, it is for the purpose of testing the applicability of welding to structures where the stresses are low but vibration is always present, the result of fatigue tests is of great importance. On the other hand, if the structure is a large tank for liquids, the fatigue test result is not as important as that of the tensile test and the bending test.

It is suggested that for general purposes the "commercial standard" be evaluated by weighing the tensile and fatigue tests by 1 and the bending test by 2.

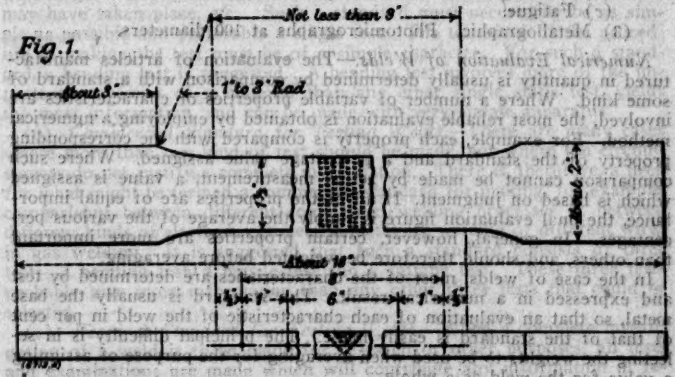
STANDARD PROCEDURE IN MECHANICAL TESTS OF WELDS.

Of the three kinds of tests which are made for the purpose of investigating a weld, mechanical tests are the most important and are in the greatest need of standardization, because they yield information about those characteristics which are of greatest significance in the majority of applications of welding. Furthermore, they are the most easily made, are better understood by a larger proportion of those interested in welding, and the results are more readily interpreted.

There are five general classes of mechanical tests which may be made on ordinary structural materials—tensile, bending (transverse), torsion, impact and fatigue. All five tests may be made on specimens cut from a sample weld, but tensile, bending and fatigue tests will give ample data for all ordinary purposes. The impact test undoubtedly has much value in connection with certain applications of welding, but it is questionable if a static torsion test develops any information that is not obtained from a tensile and a bending test. A standardized procedure is therefore particularly needed for tensile, bending and fatigue tests. The procedures outlined in the following paragraphs have been used in a large amount of

STANDARD TEST SPECIMEN IN AMERICA FOR ROLLED OR SHEET MATERIAL

Fig. 1.



testing work and have been discussed to some extent for standards. It is believed by the author that standardization along the general lines indicated will be in the right direction.

Tensile Tests.—Standardization along new lines should utilize existing standards as far as possible. The standard test specimen in America for rolled or sheet material is that of the American Society for Testing Materials, and is shown in Figure 1. It has been suggested that tensile specimens from test welds* be of the same shape and dimensions.

In some kinds of welding the test weld will not be flat, and in such cases the wide sides of tensile specimens should be at least rough-machined in order to insure parallel sides. In any case, excess metal at the weld should be removed, so that the cross-section area throughout the reduced section is constant.

Gauge marks should be laid off along the center lines of both edges 1 inch apart as shown in Fig. 1, the weld to be at the center of the middle inch. The corresponding points on the edges should be in the same straight line perpendicular to the lengthwise axis of the specimen.

It is particularly necessary in testing weld specimens that the stress be applied in a straight line along the axis of the specimen, because any bending action due to non-alignment of the grips may affect the results, particularly if the weld is brittle. The rate of movement of the crosshead of the testing machine should not exceed that at which the beam can be kept accurately balanced at all times. The observations should include the load at the yield-point and at failure, the former to be taken by the "drop of the beam" method or with dividers.

The results should include the yield-point and ultimate strength expressed in pounds per square inch of the original area; total elongation after rupture in 1 inch, 2 inches and 8 inches expressed in per cent., each of the measured results as well as the average of corresponding results to be recorded; contraction of area in per cent. The usual notes on the character of the fracture should be recorded, including comments on the extent of gas pockets, slag inclusions, or incomplete fusion which may be shown.

Bending Tests.—Of all the tests that are made on welds, none requires standardization more than the bending test, because it is of an arbitrary character, and there is therefore a wide divergence in the details of procedure. The result of a bending test is usually expressed as the angle through which the specimen can be bent when the first fracture appears, but this angle is obviously greatly affected by:—

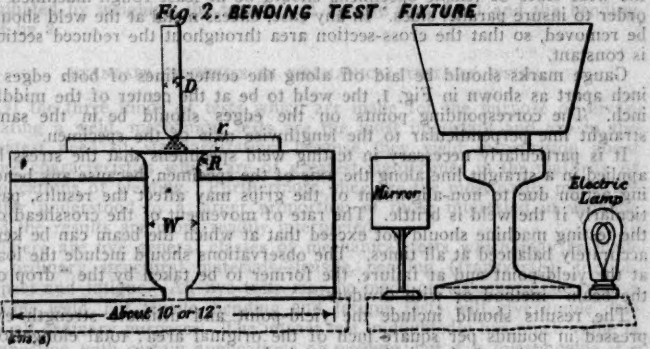
(*a*) The radius of curvature of the supports.
(*b*) The shape and dimensions of the loading surface.
(*c*) The distance between the supports.
(*d*) The method used in measuring the angle.

Anyone who has made bending tests of welds will agree that all of these conditions influence the results, and where they differ it is impossible to make comparisons which have any value. Reference to results of tests of base or parent metal obtained with the same apparatus does not eliminate the difficulty, because many parent metals—mild steel, for example—will bend to the limit of 180 degrees over a wide range of the conditions enumerated above. It is, therefore, absolutely necessary to have a carefully standardized bending test procedure before bending tests of welds can be compared.

* A term suggested for a sample or trial weld which is to be cut into specimens for testing purposes.

The bending test fixture shown diagrammatically in Fig. 2 has proved satisfactory, and is one in which the features mentioned above can be readily standardized. It will be seen that the specimen is tested as a simple beam with a minimum distance between the supports and with the weld at the center, the load being applied at the center of the weld. Such an arrangement assures a severe test of the weld, because the joint is a maximum part of that portion of the specimen which is between the supports. The fixture is conveniently constructed if two pieces of steel rail

Fig. 2. BENDING TEST FIXTURE.



are used for the supports, with suitable provision for adjusting the spacing to accommodate various thicknesses of specimens. Fig. 3 is a photograph of such a fixture as used at the Bureau of Standards at Washington.

The following are the essential dimensions:—

D = diameter of cylindrical loading surface = thickness of test specimen *t*.

R = radius of corners of supports = 1 inch.

W = distance between supports = D + 2 *t* + 1/8 inch.

The top surfaces and corners of the supports are polished and lubricated with a heavy lubricant. In the case of a weld made with metal deposited in a "V," the specimen is always tested with the load applied to the side of the specimen which shows the least filled-in metal. The fixture can be conveniently used with a standard testing machine of the vertical type, but a simple, self-contained structure for applying the load can be readily constructed.

The quantity measured in the test is the total angle through which the specimen is bent at the instant the first crack appears in the specimen. As the test is usually made with the load applied in a vertical direction and on top of the specimen, it will be found convenient to view the under surface of the specimen by means of a mirror and an electric lamp suitably located as indicated in Fig. 2.

The angle of bend can be quickly measured by the simple method shown in Fig. 3. The outline of the upper edge of the specimen is marked on a sheet of paper held against the edge of the specimen (before the load is

removed), and the angle measured with an ordinary protractor. As the specimen may not have been perfectly straight initially, a similar procedure should be followed before the load is applied. The difference between the two angles is the angle of bend which the specimen withstood, and is compared with the similar angle for a similar test of a specimen of the base metal. The width and length of the specimen may be any convenient values as far as the test itself is concerned. A width of 1.5 inches and a length of 6 inches have been found convenient for ordinary thicknesses of material. It is obvious that if the test is made with an ordinary testing machine, the fibre stress can be calculated by the standard flexure formula for a simple beam supported at the ends and loaded at the center.

Fatigue Tests—The fatigue test is of great importance in an investigation of welds, particularly those which are to be used in structures in which rapidly varying stresses exist, such as ship hulls, building structures and machinery. It has long been recognized that the usual static tests do not necessarily furnish information from which the performance of a material under a long-continued variable stress can be predicted. It is desirable to know the effect of any type of joint in a structural member on the fatigue strength of that member. With a welded joint, the question becomes particularly important because of the introduction of more or less metal of a different character.

A fatigue test, like a bending test, is largely arbitrary. That is to say, the result obtained is an abstract one which has significance only when compared with other results obtained in exactly the same manner. In the case of the tests of welds, the final answer is the ratio of the result of the test of a specimen containing the weld to that of a similar specimen of the base metal. Consequently, the exact method of procedure employed may not be as important as in the bending test, for example, provided the same test is made on both weld specimens, and base-metal specimens. However, experience may show that this ratio is not reliable, for example, over a wide range of thicknesses of material, so that standardization of fatigue test may be found to be necessary quite apart from the fact that it would be desirable on general grounds.

Many methods have been proposed for testing materials under varying stresses, and a number of machines have been developed commercially for making such tests of the usual structural materials, but the majority of these machines develop the maximum stress at a single cross-sectional plane of the specimen. A welded joint is not a homogeneous structure, for it not only consists of different materials, but in the case of a "V" weld, the area of cross-section of the filled-in metal varies throughout the joint because of the V-shape of the filled-in metal and the irregularity of the line of union between the filled-in metal and the original metal. It is highly desirable, therefore, that the type of fatigue test employed for welds be one in which a uniform stress is developed throughout that part of the test specimen in which the weld is located.

A fatigue testing machine in which this condition exists was developed for the Research Sub-Committee of the Welding Committee of the Emergency Fleet Corporation. It was described by the author in a paper recently presented before the American Society for Testing Materials.*

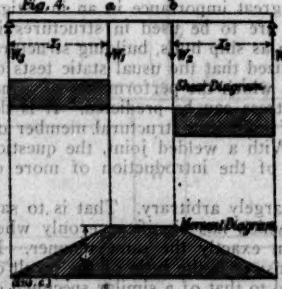
The principle employed is that of a four-point rotating beam with two free supports at the ends and two loads at the middle. Fig. 4 shows the

* "A Fatigue Testing Machine." P. M. Farmer, Am. Soc. for Testing Materials. Annual Meeting, June, 1919.

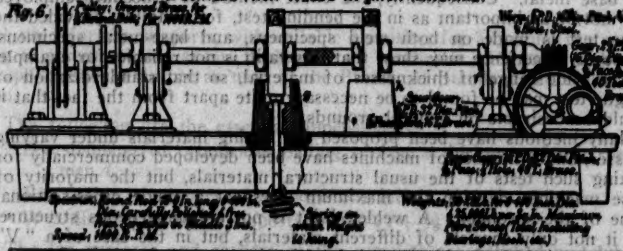
familiar shear and moment diagrams for a beam so loaded, in which $W_1 = W_2 = W_3 = W_4 = W$ and $X_1 = X_2 = X$. The moment between a and b is uniform, and has the maximum value of W X. If the beam has a uniform circular section, the maximum fibre stress in all sections between a and b is:

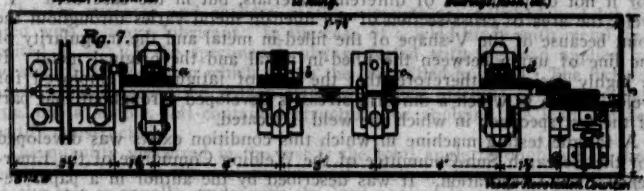
$$S = \frac{Mc}{I} = \frac{W X}{0.0982\, d^3}$$

in which S = stress in outer fibre in pounds per square inch, M = bending moment in inch-pounds, c = distance from neutral axis to outer fibre



MACHINE FOR SPECIMENS FROM WELDS IN 1-INCH MATERIAL.





in inches ($= d/2$), I = moment in inertia, W = load in pounds, X = distance between loads and supports in inches, and d = diameter of beam in inches. The beam is in tension on the lower side and in compression on the upper side. Thus if a circular rod is turned from a specimen cut from

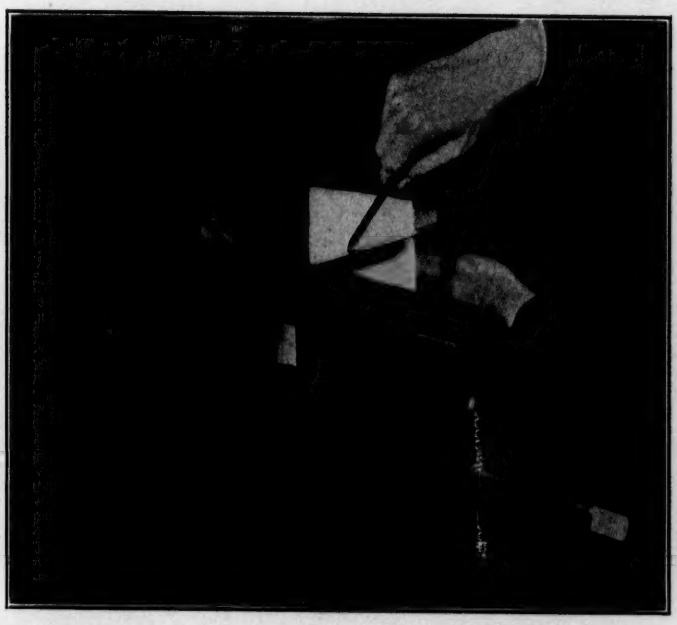


FIG. 3.—BENDING-TEST FIXTURE AT THE BUREAU OF STANDARDS, WASHINGTON.

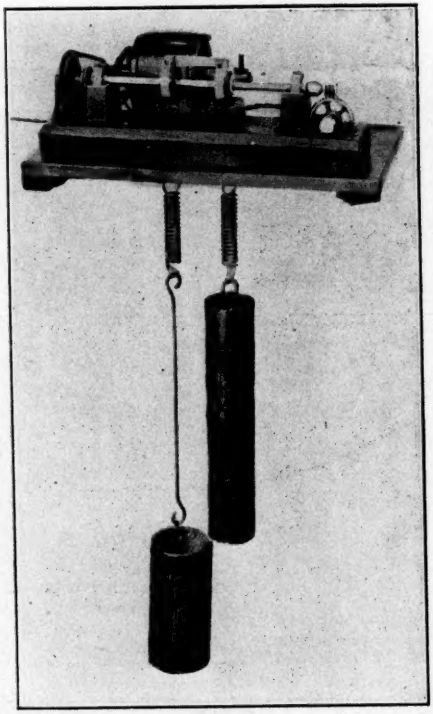


FIG. 5.—MACHINE FOR SPECIMENS FROM WELDS IN ½-INCH MATERIAL.

a weld, loaded in the manner indicated above with the joint in the middle and then rotated, the joint will be subjected to a maximum stress equal to S alternately in tension and compression once per revolution.

Fig. 5 is a photograph of a machine for specimens from welds in ½-inch material, and Fig. 6 is reproduced from a drawing. The specimen is carried by four self-aligning ball bearings, *a*, *b*, *c* and *d*. The two end bearings rest in the pillow blocks, *e* and *f*, the whole being supported on a cast-iron base 5 inches by 19¼ inches. One of the pillow blocks provides for longitudinal movement of the bearing, due to the slight bending of the specimen. The two equal loads are hung from hooks attached to collars which encircle the outer faces of the two middle bearings.

The bearings are securely fastened at the proper position by means of two split collets of carburized steel with opposite tapers, one entering from each side of the bearing. This produces a wedge action which gives a uniform pressure over the entire bearing surface on the shaft, and insures the validity of the assumption that the point of application of the loading forces is at the center of the bearings. The inner collet is counterbored as shown, so that the bearing surface on the specimen is as short as possible. The collets are wedged together by a nut on a thread on the inner collet, and are loosened with the aid of a wedge key driven under the head of the inner collet.

The specimen is rotated through the loose coupling at one end of the specimen, and the revolutions are counted by a suitable counter geared to a worm on the other end of the specimen. These various fittings are most conveniently put in position on the specimen at a vice on a bench. The specimen with the fittings in position is then dropped into position in the machine, the weights hung on to the hooks (with spiral springs intervening), and it is ready to be rotated. The blocks *g* and *h* are merely for the purpose of preventing the weights from dropping too far when the specimen fails.

When failure occurs, the specimen will stop rotating because it "binds," and the belt will either slip or be thrown off. However, the preferable way for securing automatic stopping is to provide a simple switch arrangement which will be operated by one of the weights when it drops. This will cut off the motor.

This particular machine was designed for specimens from welds in ½-inch plates, but obviously a machine could be constructed for specimens of any ordinary diameter. A machine could be constructed with a longer base for specimens from ⅝-inch to 1-inch plate by providing bushings of suitable sizes and arranging for changing the distance between the supporting pillow blocks, so that the weight can be kept within reasonable limits.

The standard specimen for the machine described is 13 inches long and 0.4 inch in diameter throughout its entire length. This diameter is about the maximum that can be turned out of a specimen cut cross a weld in ½-inch material which is rarely perfectly straight. Care is taken that the diameter in the middle portion where the weld is located is exact, and that this part of the specimen is carefully finished free from tool marks or scratches.

The criterion of the performance of a joint of any kind is the performance of a similar specimen of the original metal without a joint. Consequently fatigue tests of welds in ½-inch plate are compared with tests made with the same apparatus and at the same stress on specimens from the original plate. When such tests were made with specimens of constant diameter throughout their length, failure always occurred at one of the load bearings and not between them, as is the case with weld

specimens where the joint is usually the weakest part of the specimen. It is believed that this is due to the sudden change in the stress distribution, as shown in Fig. 4, at these points, and probably also to the radial compressive stress induced at these points by the loads. By reducing the diameter between these points and using long fillets, failure will occur in the central portion of the specimen and the true endurance value of the metal will be obtained.

The objection might be made that the specimen is too small to give reliable information about what a weld of any considerable length would do, unless a number of specimens were cut along the weld. In other words, the machine should be such that it would take a comparatively wide specimen. The answer is that the very fact that the small specimens require a number of tests to be made is an advantage, because definite knowledge of the uniformity of the strength (or lack of it) along a long weld is very important, and obviously this would not be obtained with a single wide specimen. Furthermore, with the much greater speed permissible with a small machine and the larger number of machines which could be available, the equivalent in small specimens can be tested just as quickly as the one large specimen. Of course, a large machine employing the principle of uniform stress over the entire weld capable of taking a long specimen is a great value in research work and has its place, but it is questionable if such a machine would be suitable or practicable as a standard for general use. A machine of this kind was designed by Professor H. F. Moore at the Bureau of Standards at Washington, D. C. It takes a flat specimen, 8 inches wide, 32 inches long, and of any thickness up to about 1 inch. The principle of the machine is exactly the same as that described above. The specimen is, of course, stationary, and the load is applied at the two loading points alternately up and down by means of an electric motor.

Number of Specimens that Should be Tested.—A property which is most important in a weld is uniformity. It is therefore desirable to subject a number of specimens to each of the various mechanical tests. A separate sample or test weld can be prepared for each specimen, but in electric arc-welding in particular this means greatly increased expense in metal, in making the welds and in machine work involved in preparing the test specimens because of the necessity for cutting away enough of the edges to eliminate the beginning and ending of the weld. In electric arc welding at least it is more economical to make one large test weld, and cut it up into the pieces from which the test specimens are machined. In gas welding a skilled workman may be capable of making a finished weld right up to the ends of the weld, in which case making each test specimen separately might be more economical. But the method of making one large weld has the further advantage that it more nearly duplicates the usual practical conditions, that is, long joints with the attendant temperature effects as determined by adjacent, relatively large masses of metal.

Not less than three specimens should be subjected to each of the mechanical tests. They should be cut from the test weld in such a manner that one specimen for each test is cut from the middle and one near each end, enough of the edges being first removed to eliminate defecting welding which may exist at the beginning and end of a weld.

Elongation Data in Tensile Tests.—Elongation measurements on weld specimens are of limited value. The specimen is not homogeneous throughout its length, and unless the ductility of the filled-in metal of the joint is the same as that of the base metal, the result obtained is a composite of the ductility of the filled-in metal and the base-metal. Such

measurements have some value, however, when comparing welds of the same class with the same base metal. In a thorough investigation of a weld of the deposited-metal type, data on the ductility of the filled-in metal itself should be obtained by making tests of specimens consisting of filled-in metal only.

Width and Gauge Length in Tensile Tests.—The use of a specimen of constant width and gauge length, irrespective of thickness, might be questioned on the ground that it disregards Barba's law of proportionality, which states that the ratio of the gauge length to the square root of the area should be a constant. Investigations made by Professor H. F. Moore, under the auspices of the Committee on "Methods of Testing" of the American Society for Testing Materials,* showed that varying values of this ratio (within reasonable limits) do not affect the yield-point, elastic limit, reduction of area, or the tensile strength (of specimens of homogeneous material), but that there is a marked effect on the elongation.

So far as elongation data in tensile tests of welds are concerned, it has already been pointed out that such data are of limited value. This being the case, it does not appear that in ordinary tests of welds the additional complication involved in following standard dimensions based on Barba's law would be justified. But where investigations are being made to determine the effect of thickness on the weld, it may very well be advisable to follow this law. In that case it would appear desirable to make the gauge length as short as possible, and such that the ratio of the filled-in metal to the base metal within the gauge marks is constant. Then the width should be varied to keep the ratio $\frac{L}{\sqrt{a}}$ a constant for various thickness. For example, if test welds were made in various thicknesses of plate with a 90-degree angle in each case, and a specimen 1.5 inches wide with a 2-inch gauge length were prepared from the 0.5-inch material, the dimension of the other specimens would be as in the following table, $\frac{L}{\sqrt{a}}$ being constant at 2.31.

Thickness.	Width.	Gauge Length
In.	In.	In.
0.125	0.375	0.50
0.25	0.75	1.00
0.50	1.50	2.00
0.75	2.25	3.00
1.00	3.00	4.00

Obviously, the tests of base metal which would be made for reference should be made on specimens of corresponding shape.

Standard Stress Value in Fatigue Tests.—In the fatigue tests made for the Research Committee of the Emergency Fleet Corporation, a standard maximum fiber stress of 25,000 pounds per square inch was employed in testing of weld specimens and the corresponding specimens of base metal. The selection of this value was based on the following considerations. The characteristic which it is desired to test is the elastic property

* Proceedings, Am. Soc. for Testing Materials, vol. xviii, Part I, page 300 (1918).

under reversed stress conditions and not ductility, since in practice a material is never intentionally stressed above the elastic limit. This means, therefore, that the stress should be below the elastic limit, because the relative performance of materials in a fatigue test will probably be very different above the elastic limit where the property involved is ductility, than it will be below the elastic limit where elasticity is the property involved. On the other hand, the higher the stress, the more quickly results will be obtained. As there is no reason to suppose that the relative behavior of different materials will not be the same at low stresses as at high stresses (provided they are below the elastic limit), a stress of 25,000 pounds was adopted. This is as close to the elastic limit of ship-plate steel as it was thought advisable to go.

In an extensive research investigation, it might be advisable to determine the stress at which failure occurs after a stated number of reversals (or repetitions) of stress by the application of Basquin's exponential law,*

$$S = \frac{K}{N^x}$$

where S = fiber stress, N = number of repetitions of stress at failure, and K and x are experimentally determined constants. Tests would be made at a number of different stresses below the elastic limit and the results plotted on logarithmic paper. According to this law the curve should be a straight one, so that the stress which will cause failure at any particular number of repetitions of stress is readily determined.

This method gives a more fundamental basis of comparison. It permits direct comparison of the fatigue strength of different kinds of welds and welds in different materials without reference to the fatigue strength of the base metal. It is obvious, however, that the various specimens must be very uniform if reliable information is to be obtained with a reasonable number of tests.

Inspection and Tests of Welds in Structures.—The phase of the welding art which is in greatest need of standardization is that of inspection and testing of completed welds in commercial applications. But progress towards this desirable end can hardly be expected before a satisfactory and commercially practicable method of determining the quality of a weld in a structure is devised. Investigations of various suggested methods of testing completed welds were made by the Welding Committee of the Emergency Fleet Corporation. These included X-ray examination, determination of the electrical resistance by a method somewhat similar to that employed in testing rail-joints in electric railways, hammering and chipping the weld and magnetic examination. None of these offered definite promise of success at the termination of the committee's activities. In certain limited fields of application, satisfactory tests may be devised, such, for example, as closed vessels which can be subjected to hydraulic pressure and simultaneous hammering after completion. Possibly also some method will be devised for removing a test-specimen which can be readily tested, the hole being small enough to be filled up easily.

But at the present time the only checks on the quality of the welder's work in general applications which are available appear to be (*a*) confidence in the welder's proficiency and his integrity, and (*b*) systematic inspection during the progress of the work. Even here standardization may be applied advantageously. That is, we should have (*a*) a standardiza-

*O. H. Basquin, "The Exponential Law of Endurance Tests," Proceedings, Am. Soc. for Testing Materials, vol. x, page 625 (1910).

tion test of proficiency of welders or perhaps several standards for different grades of welders and (*b*) a standardized procedure for the inspection of various kinds of welds while being made. Standardization of this kind should have the added value of aiding in obtaining the confidence of those authorities whose approval is necessary before welding can be applied in a number of important fields.—"Engineering," Feb. 25, 1921.

STEAM TURBINE DEVELOPMENT.

A study of contemporary technical literature reveals the fact that a very large amount of attention is being paid to the subject of the marine Diesel engine; we find, further, that those learned societies which deal with marine engineering and kindred problems are receiving many papers on the various phases of the same subject. On the other hand, a study of the survey of mercantile shipbuilding in 1920, recently issued by the Committee of Lloyd's Register (see "Shipbuilding and Shipping Record," January 27, page 104, *et seq.*) shows that out of a total of 618 vessels representing 2,055,624 tons of shipping launched in the United Kingdom during 1920, only 25 vessels of 86,940 tons were to be propelled by oil engines, as against 566 vessels, representing 1,953,014 tons, to be propelled by steam engines. That is to say, 95 per cent of the total tonnage launched was to be driven by steam against 4.2 per cent by motor engines. Of this remarkably high percentage of steamers, 79 vessels with a total tonnage of 638,557 tons were to be fitted with turbines, this being approximately 31 per cent of the total tonnage launched during the year, although it is only 13 per cent of the number of vessels launched. This apparent discrepancy is due to the fact that, generally speaking, it is the larger steamers which are propelled by turbines, all the vessels of 14,000 tons and above being thus propelled. It is apparent, therefore, that the internal-combustion engine is receiving a far larger amount of expert attention than is warranted by the extent of its actual adoption for merchant ship propulsion, and while this is more or less understandable in view of the potentialities of the oil engine, it can hardly be denied that the steam engine in its two different forms is in danger of being neglected, and any possibilities of further development are likely to be jeopardized. The reciprocating steam engine has, of course, more or less reached a state of finality, although there are still certain directions in which improvements might be made even with this form, but the marine steam turbine is still, comparatively speaking, in its infancy, and there are a large number of problems in connection with the design of the turbine itself, the necessary auxiliaries and the speed reduction gearing which demand a solution.

During the past year it has been noticeable that the reaction type of turbine is again coming into favor. This does not mean that the impulse type has been found wanting, nor that we are reverting to the early type of multi-stage reaction turbine such as was adopted in the first days of the application of the turbine to ship propulsion, and in which the reaction principle was employed throughout the whole expansion of the steam. But it appears as though, with one or two notable exceptions, designers prefer to utilize the reaction blading at the lower pressures employing a single impulse wheel at the high pressure end. The advantages that accrue from the adoption of this arrangement are many, not the least of which lies in the fact that a considerable fall of temperature occurs at the impulse wheel, and, therefore, the range of temperature throughout the remainder

of the turbine is greatly reduced, thus minimizing the effects of distortion of the casing and of the rotor due to the fluctuations in the steam pressure or in the degree of superheat. There are, on the other hand, corresponding disadvantages due to the increase in the number of stages required and to the finer clearances demanded by the reaction blading as compared with those in the impulse wheels, and it still remains an open question whether the pure impulse turbine is superior to the combined impulse-reaction type. There is also ample scope for investigation and research in the direction of finding suitable means of resisting the distortion and the erosion which occurs when highly superheated steam is employed. The gain in economy which follows the use of highly superheated steam is undoubted, but against this has to be set the more rapid deterioration of the engine itself, and it is therefore essential that some simple means should be devised whereby the effects of high temperature steam moving with high velocity can be overcome if we are to take full advantage of the higher efficiency which the use of superheated steam renders possible.

The mechanical reduction gearing has so far thoroughly justified the high opinions which were expressed regarding it in the early days of its adoption. There are, nevertheless, certain features in the design and construction of gears which are amenable to improvement. We have yet to understand why it is that pitting of the wheel teeth in the region of the pitch line is found to occur after the first voyage, and while happily this does not as a rule grow worse subsequently, it introduces an uncertainty into the problem which marine engineers would rather have eliminated. Again, the question of single *versus* double reduction gearing is one which calls for the closest investigation, and we have little or no data as to the behavior of gears transmitting large powers. The success which has been achieved on H. M. S. *Hood*, where 140,000 horsepower is developed at four shafts, necessitating the employment of gears transmitting 35,000 horsepower each, is highly satisfactory, and indicates that the mechanical reduction gearing can be safely installed on the largest and fastest vessels; but the occasional records of failure, and the lack of consistency in the behavior of similar gears of the same size and make, all tend to prove that there are certain factors in the problem of the design of mechanical reduction gearing which are not yet thoroughly understood.

We have briefly hinted at one or two of the outstanding problems which face the designer of the marine turbine and the designer of the speed reduction gearing, with the object of dissipating the mistaken but prevalent idea that the marine geared turbine has already become standardized. Standardization at all times is dangerous since it tends to produce stagnation, and nothing is more harmful in any branch of engineering than the absence of the desire to improve. The steam turbine has shown itself to be a remarkably efficient form of prime mover for the propulsion of ships, particularly when used in conjunction with speed reduction gearing, but it is still capable of considerable improvement. The day seems very far distant when the internal-combustion engine will displace the steam turbine, particularly for the development of large powers; but there are still many problems in connection with the design and operation of the latter type of engine which require elucidation. The turbine is still capable of being considerably improved, as also is the gearing, and it will only be by careful investigation and research that the highest possible degree of efficiency and reliability will be achieved.—"Shipbuilding and Shipping Record," March 3, 1921.

THE INTERNAL-COMBUSTION TURBINE.

The late Lord Fisher was one of the staunchest advocates of the internal-combustion engine for the propulsion of naval vessels, and he declared on more than one occasion that the problem of designing a satisfactory engine for large units was a metallurgical one. It is generally recognized that the true path of development of engines working on this principle lies in the production of the internal-combustion turbine, and it is here far more than in the reciprocating gas or oil engine that the metallurgical problem enters. The marine Diesel engine has been brought to a considerable pitch of perfection, although designers do not fail to recognize that it still suffers from certain limitations, notable among which is the comparatively small power that can be developed per cylinder. We need not enter here into the reasons why cylinders of larger diameter cannot be employed, suffice it to say that they are mainly due to our inability to produce a metal which can withstand the effects of the high temperatures inseparable from the efficient operation of the Diesel cycle, so that here also the problem to be met by designers is mainly a metallurgical one. Lord Fisher was credited with having the oil turbine in mind when referring to the ideal type of engine for naval vessels, and in view of his strenuous advocacy of the steam turbine in the early days of its development, his faith in the possibilities of the rotary internal-combustion engine can be well understood. His almost prophetic insight is, however, more than justified when one studies the details of the new "Holzwarth" oil turbine, particulars of which have recently been published, for it is apparent that the design has been hampered to a very great degree by the knowledge that there is not as yet a metal which can successfully withstand for any considerable time the effect of high temperatures.

The machine under notice is now undergoing tests in the works of the manufacturers, Thyssen & Co., at Mulheim, Germany, and it is claimed that it is the first commercial oil turbine in the world. It develops 500 b.h.p. at a running speed of 3,000 r.p.m., and while this particular unit has been built to drive an electrical generator, it is understood that an exactly similar machine is to be built with the idea of testing it on board a ship. The principle adopted for the cycle of operations follows closely along the lines which have been utilized by earlier experimenters, the necessary compression being obtained by a separate machine which in this instance is driven by a small steam turbine, the steam being obtained from a boiler heated by the exhaust gases. Air from the compressor is led into a combustion chamber forming an integral part of the turbine, into which the oil vapor is forced, the burnt gases passing through a series of nozzles, and impinging at high velocity on to the blades affixed to the rotor. The most notable difference between this particular machine and the majority of those which have preceded it lies in the greatly reduced compression pressure that is employed, this being so low that the temperature generated is not sufficient to cause ignition of the fuel at the point of injection and in consequence an electric spark has to be employed. This means that not only is there the extra complication and expenditure involved in the provision of sparking gear, but that the turbine can only be run on the more easily vaporized oils such as petrol and paraffin.

But of even greater importance than the above is the fact that by the limitation of the compression pressure the thermal efficiency of the engine must be seriously affected. It can be readily demonstrated that the thermal efficiency of any heat engine depends upon the ratio of compression, and it is one of the greatest features of the Diesel engine that by employing a high ratio of compression a high thermal efficiency is obtained; the ad-

vantage of automatic ignition of the fuel at the moment of admission follows as a result of the high compression. Undoubtedly the necessity of keeping the temperature throughout the cycle within manageable limits is responsible for the low compression that is employed in this turbine, and when it is stated that the explosion pressure is only about 205 pounds per square inch, it will be realized how great is the sacrifice of efficiency involved. The designers recognize this, for they attempt to improve the performance of the turbine by utilizing the exhaust to generate steam for driving the compressor, which while it may be advantageous from the point of view of fuel consumption, must add to the weight of the entire machinery and to the space it occupies, two serious disadvantages from the point of view of the adaptability of this form of oil turbine for marine work. The results achieved, however, at this low compression pressure must be considered highly successful. It is claimed that the turbine blades operate very satisfactorily at the temperatures to which they are subjected, and it is well known that the construction of the blades has hitherto been one of the most difficult problems in the design of internal-combustion turbines. These blades are made of electric steel specially heat-treated.

It must be admitted that hopeful as the prospects of this particular design appear, we are still very far from having solved the problem of the oil turbine. In the first place what is wanted is an engine which will run on heavy residual oil as does the marine Diesel engine at the present time; moreover the compression must be sufficiently high to ensure a thermal efficiency as high if not higher than that obtained in existing types of oil engines. To this end the compression must be considerably greater than that employed in the turbine under discussion, in fact, high enough to ensure automatic ignition on injection; although possibly the adoption of a device similar to that employed in the semi-Diesel engine might be adopted in order to limit the compression pressure if this is found to be absolutely unavoidable. Further, the idea of installing an entirely separate air compressor appears to us to be unsound, and failing some arrangement equivalent to the return stroke of the piston it is essential that the compressor should form an integral part of the turbine, delivering its charge direct to the combustion chamber. In conclusion, if higher pressures are employed, it follows that the material of the blades and, to a less extent, the whole of the parts subjected to contact with the products of combustion, must be of such a nature that it can withstand for long periods the erosive action of the gases. This indeed is the crux of the whole problem, and when the requisite material is forthcoming the main obstacle in the path of progress towards the design of the marine oil turbine will have been surmounted.—"Shipbuilding and Shipping Record," Feb. 24, 1921.

AMERICAN DESTROYER HITS MINE IN THE BALTIC.

By Commander E. S. Land, C. C., U. S. N.*

The peace has its naval difficulties as well as war is typified by the accident to the U. S. S. *Kane*, on October 1, 1920, when this American destroyer hit a floating mine in the Baltic just outside the Gulf of Riga. The *Kane* was well clear of the mine fields and no difficulties with regard to mines were expected, although an extremely careful watch was kept, as

* Naval attaché, American Embassy, London.

is customary in these waters. Fortunately, there were no casualties to the personnel, although the violence of the explosion caused minor injuries to three of the crew as they were hurled against bulkheads and stanchions.

The remarkable part of the damage to the *Kane* is that most of the force of the explosion took effect at and above the waterline. To attempt to follow out the relations of cause and effect in a mine explosion is a good deal like deciphering Egyptian hieroglyphics. In this connection, attention is particularly invited to the damage to the 4-inch gun mount.

PRINCIPAL DIMENSIONS OF THE VESSEL.

The *Kane* is 314 feet long overall, with 176 frames spaced 21 inches apart. Her beam is 39 feet 10½ inches; draft, 9 feet 4 inches. and her full load displacement, 1,300 tons.

The explosion occurred off the port quarter. There is strong circumstantial evidence that one of the port propeller blades struck a mine floating just below the surface, as the leading edge of the blade was crushed to a depth of from ¼ to ½ inch, for a distance of 14 to 18 inches.

The after end of the ship was apparently bent bodily to starboard, bending showing up throughout the girth of the ship from frames 141 to 143, i. e., about 65 feet from the stern of the vessel. This bending included, to a greater or less degree, both line of shafting, particularly the port shafting, as the port engine could be turned over only with much chattering of the gears and heavy vibration of the ship. The starboard engine could be turned over up to one hundred revolutions per minute, but this caused considerable chattering of the gears and very marked vibration of the ship. This springing of the shafts occurred in the tail shafts; the port stern shaft (the section of the shaft just forward of the tail shaft) was bent.

The bearings and couplings connecting the engine shaft and stern shaft were not damaged. The steering gear and rudder were first thought to be badly damaged, but afterwards found to function reasonably satisfactorily. The main and auxiliary machinery were not damaged to any appreciable extent.

STRUCTURAL DAMAGE.

The main deck starboard stringer was buckled to a depth of four inches between frames 141 and 142; the stringer angle was broken between frames 141 and 142; the buckling was carried across the deck between frames 142 and 143; the main deck stringer plate, on the port side, was buckled at frame 142, the depth of the buckle on the port side was about 2½ inches. The starboard stringer plate was torn for about 2 inches at the outboard edge and one rivet was gone. This was the only tear in the plating.

The starboard side plating was buckled at the top strake between frames 141 and 142, running diagonally aft to 143 on the waterline. The frames, brackets, and bounding bars were buckled and bent inboard of the buckled plating from the main deck to the garboard strake. One air port lens was broken and its frame distorted. The deck plating adjoining the port stringer plate had hair cracks in it; otherwise there were no cracks or holes in any of the deck plating.

The side plating on the port side had a buckle about 1½ inches deep in the top strake of the plating between frames 142 and 143; the two top strakes were also badly buckled from frame 165 to 172, the third strake of

plating from the top was slightly buckled. The stringer plate on top was bent and buckled slightly, from frames 165 to 172. This buckling between frames 141 and 143 occurred in the decks and steel structure inside the ship as well as in the side plating and top side.

PROPELLER GUARDS.

Three after sections of the port propeller guard were gone and two forward sections were damaged. The second strake of plating from the main deck was slightly buckled over the port strut.

CREW SPACE.

Compartment D-204 on the first platform deck was buckled all the way across just forward of frame 142, to a depth of about 3 inches. There were small buckles at frame 144 on the starboard side only.

MAGAZINES.

Compartment D-108-M sprung a small leak on the port side between frames 141 and 142 directly under the shaft. The bulkhead between D-107-M and D-108-M buckled between frames 141 and 142. Compartment D-107-M sprung a leak between frames 141 and 142, directly under the shaft as well as in the garboard strake on the starboard side between frames 141 and 142; this strake was buckled slightly to a depth of from 1 to 2 inches. Compartment D-109-M had its side plating badly buckled between frames 143 and 144; a small hole was torn in the side plating abreast of the shafting between frames 143 and 144; the longitudinal was buckled. This leak was stopped by means of a life preserver wedged in.

STOREROOMS.

Compartments D-105 and D-106 had the spring bearings sprung on both port and starboard sides and were leaking badly at first, but the leaks were stopped by setting up on the glands. Compartment D-110 had slight leaks at the seams abreast the port shafts at frames 146 and 147. The plating on the port side of compartment D-111 was buckled from frames 159 to 161. The stanchion in this compartment was driven down slightly, although it was not buckled. The web plating, brackets, and longitudinal were buckled from frames 158 to 164 on the port side. The starboard side was O. K. The middle line frames 162 and 163, and the web brackets buckled on the port side. The peak tank, bulkhead 164 was buckled and leaked slightly at the top of the vertical keel. This leak was calked.

STEERING GEAR.

In compartment D-208, frames 164 to 172, the longitudinal and side plating were buckled. The knee brackets to the main deck were buckled and the rivets sheared, particularly at frames 168, 169, 170 and 171 on the port side. The diagonal stanchion brace on the port side at frame 168 was driven through the main deck, and the stanchion at frame 170 on the port side dropped out. The galvanized rack for oil cans on the port side of this compartment was blown adrift. A number of scattered rivets in the frames and longitudinal (fourteen in all) were sheared between frames 168 and 172. The starboard side of this compartment was O. K.

The peak tank on the port side, plating, frames and partial bulkheads were all slightly buckled or bent from frames 164 to 175. A number of leaky seams were found here.

One life line stanchion and all life lines on the port side aft were blown away. The flagstaff, steering room ventilator and one 50-gallon steel drum full of gasoline were blown overboard.

DAMAGE TO GUN MOUNT.

No. 4, 5-inch, 51 caliber gun was not injured. The gun carriage was spread apart about three-quarters of an inch at the extreme end of the trunnions, this left the gun resting on the lower cap squares of the carriage. The upper cap was not injured, but the left one was raised and split, thereby stripping the holding down bolts which secured both squares. The roller path of the training gear was damaged to such an extent that the gun could not be trained through more than six degrees. This was caused by the gun carriage and slide being driven aft, while the inside mount remained stationary. This caused the after guide lug with its holding down bolts to be broken off. The firing circuit of the gun leading from the transfer switch to the firing key, then to the pin, then to the bolts and screws holding the firing circuit was torn loose.

The tompion was in place when the explosion occurred, which complicated any efforts to explain these damages.

PROPELLER STRUTS BROKEN.

Both port propeller struts were broken, the lower one being broken at the outboard end and cracked on the inboard end. The upper starboard propeller strut was broken.

Both tail shafts were sprung and the port stern shaft was bent. Practically the complete girth of the ship except the keel was buckled at about frame 142. The stern was bent 1 inch to the starboard, commencing at frame 141.

The main keel was not damaged.

Every effort was made by the officers and men to bring the *Kane* from the scene of the explosion to Copenhagen in the most satisfactory condition practicable. The vessel was towed to Libau and from Libau to Copenhagen by the United States destroyer *Brooks*. There was considerable working of the stern of the *Kane* during this voyage, which was relieved to some extent, by fore and aft shoring across the buckled portion of the ship. On October 7, the *Kane* was docked at Landskrona, Sweden, and repairs immediately undertaken.—"Marine Engineering," Feb., 1921.

MAIN ENGINE CLUTCHES FOR GERMAN SUBMARINES.

A general inspection of surrendered German submarines has indicated that engines for these vessels were of four standard sizes. A 10-cylinder 3,000 horsepower engine was built for installation in large cruiser submarines. A number of these engines were found in the shops ready for installation at the time of the signing of the armistice, but only one pair was installed in a boat, the U-142, which performed no active war service. Six-cylinder 1,750-horsepower engines, similar in cylinder dimensions to the 3,000-horsepower engine, were installed in the U-135, 136, 139, 140

and 141. A large number of 1,200-horsepower six-cylinder engines were installed in operating submarines and gave excellent service throughout the war. A fourth size, of 530 horsepower, six cylinders, was found in late types of mine-laying and coastal boats.

The clutching device for connecting the engine and motor shafts is of special interest in view of the history of such devices in general and has led to numerous inquiries as to its construction. The same type of clutch was installed with each size of engine.

The clutch is of the double-cone friction type, of semi-steel and cast iron, operated by air except in the 530-horsepower size, which is operated by hand through worm and gear.

The outer casing or female member, which serves as the flywheel, is carried on the engine shaft. This casing, as shown in Fig. 1, is made in two parts bolted together with the inner surfaces machined to cone surfaces, the cones base to base.

The male part is attached to the motor shaft by three keys and a retaining nut, carrying a flange or spider in which are fixed eight large guide pins parallel to the shaft axis and extending through the spider on either side as shown in Fig. 1. Two inner clutch cones are carried on these guide pins, on which they have a sliding fit. They are semi-steel and their outer or peripheral surfaces are finished to conform to the inner conical surfaces of the female or flywheel part.

In Fig. 2 are detail sketches of the spider and in Fig. 3 are details of the two male cones carried by the spider. These cones are alike except that stops are cast on the after cone (the cone nearer the motor) to limit the travel of the operating collar, as shown at X in Fig. 3.

The machined surfaces of the inner cones, Fig. 3, have oil grooves which insure that when unclutched or disengaged a complete film of oil will fill the clearances between the cone surfaces. The gradual squeezing out of the oil film as the surfaces are forced together enables the clutch to be thrown in when the engine is running.

The clutch is thrown in by forcing the inner cones apart until the male and female cone surfaces engage; and by drawing the cones together and thus releasing the surfaces the clutch is thrown out. The inner cones are drawn together or forced apart by a series of links and toggles actuated by a sleeve (A in Fig. 1) sliding on the motor shaft.

Referring to the lower half of Fig. 1, the center x is shown fixed to the forward cone and center z to the after cone. The clutch is engaged by separating centers x and z by forcing center y down to a line joining centers x and z. An adjusting cam around center x provides adjustment for wear in cones and bushings of the operating gear.

It will be noted that the forward link member B joining centers y and z (Fig. 1) is a steel leaf spring and that a further downward movement of center y against the stop n tends to decompress the spring slightly and to lock the mechanism. This spring is an important element in the mechanism. It is of chrome-vanadium steel with a tensile strength (annealed) of 80,000 pounds per square inch and an elastic limit of 50,000 pounds per square inch (annealed). It is heat-treated and quenched in oil.

There are four of these sets of operating links equally spaced and placed between pairs of pins in the blank spaces of the spider, Fig. 2.

Fig. 4 shows the clutch in diagram for the 1,200 and 1,750-horsepower sizes, noting the principal dimensions, as also does the following table. No data for the 530-horsepower and 3,000-horsepower clutches are now available.

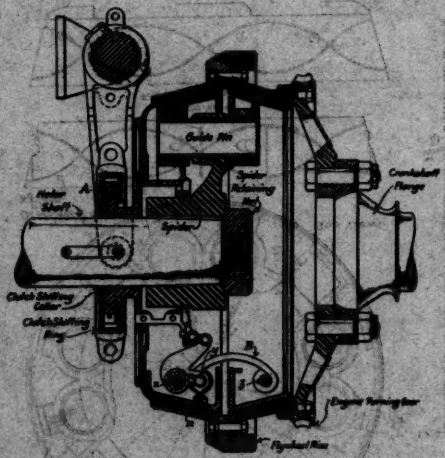


FIG. 1 SECTIONAL VIEW OF CLUTCH

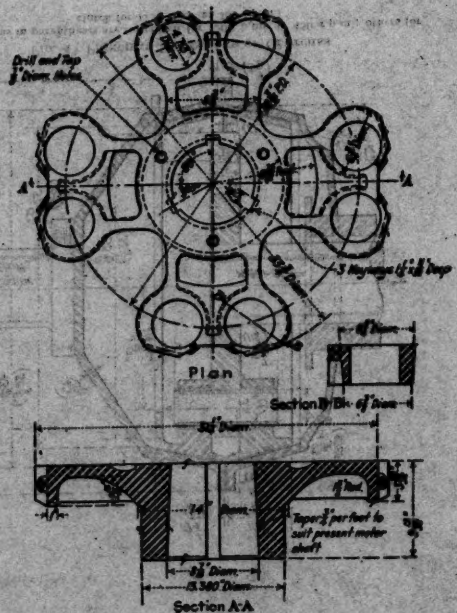


FIG. 2 CLUTCH SPIDER

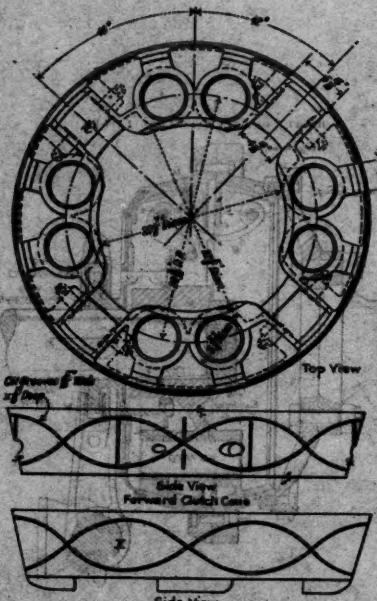


FIG. 3 DETAILS OF CLUTCH CONES

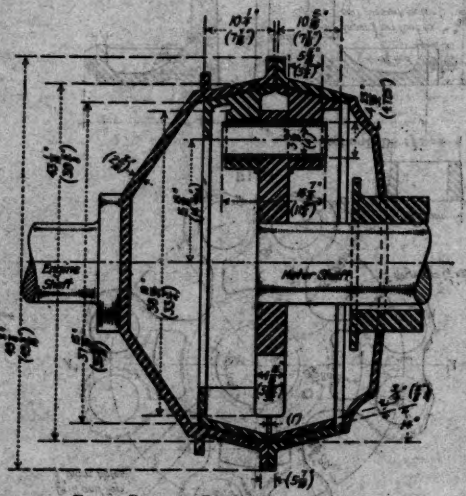


FIG. 4 PRINCIPAL DIMENSIONS OF CLUTCHES

Dimensions in parentheses are for clutch for 1200 b.hp. at 450 r.p.m.; others for clutch for 1750 b.hp. at 300 r.p.m.)

Horse-power of engine	R.p.m.	Largest diam. revolving part, in.	Cone angle, deg.	Cone diameter in max.	Cone diameter in min.	Area of friction surface, sq. in.	Crank-shaft diam. in.
1200	450	49-3/16	14	39-5/8	35-15/16	1765.112	9.453
1750	380	49-3/8	14	43-1/16	37-15/16	2668.297	11.219

The operating gear of the clutch used with the 1,200-horsepower engine consists of a double-acting air cylinder mounted on the floor just inboard of the clutch and connected to the ship's air line under 100 pounds pressure. The piston of the air cylinder drives a crosshead to which is attached a lever, the other end of which is keyed to a shaft supported by a fixed bracket above the clutch. The shaft leads over the clutch and drives a lever leading downward and ending in a fork end or yoke attached to a ring around the sliding collar which operates the clutch.

Assuming that the pressure of the air cylinder builds up to 100 pounds per square inch before the operation of clutching is complete and that maximum normal thrust in the cones is obtained when center y is 1 degree above the line of centers x and z (Fig. 1), the following data are found:

Area of operating cylinder = 18.665 square inches.
Force on crosshead = 100 × 18.665 = 1866.5 pounds.
Force on operating collar through levers = 3732 pounds.
Resolved force at center y normal to line of centers, = 1925.2 pounds, which produces an axial thrust on each cone of 1925.2/ tan 1 degree = 110,012 pounds.
Normal thrust = 110,012 × sin 14 degrees = 26,621.9 pounds.
Total normal thrust per cone = 4 × 26,621.9 = 106,448 pounds.
Friction area of each cone = 882 square inches.
Unit pressure = 106,448/882 = 120.7 pounds.

Referring to the engine itself,

B.h.p. of engine = 1200; r.p.m. of engine = 450.
K = coefficient of fluctuation of turning moment = 1.6 for 6-cylinder engine.
M = maximum turning moment of engine in inch-pounds =

$$\frac{1200 \times 63{,}000 \times 1.6}{450} = 168{,}000 \text{ inch-pounds} \times 1.6 = 268{,}800 \text{ inch-pounds.}$$

Assuming now that

N = total normal pressure on cones; R = mean cone radius
f = coefficient of friction; $N = M/fr$; $f = M/Nr$

then $f = \frac{268{,}800}{106{,}488 \times 18.89} = 0.133$ (coefficient of friction for steel on iron, freely lubricated).—Extract from "Mechanical Engineering," April, 1921.

ONE-PIECE GUN CONSTRUCTION.

In a paper recently read before the American Institute of Mining and Metallurgy, Professor P. W. Bridgman, of Harvard, who is famous for his experiments on water and other bodies under extremely high pressures, described an experiment in one-piece gun construction carried out under his direction by the Naval Authorities of the United States. The gun in

question had an internal diameter of 3 inches; it was 23 calibers in length, and was made from a single forging. Were the mathematical theory of elasticity directly applicable to gun stresses such a single piece gun should be very much weaker than a built-up or wire wound gun, but as is well known the elastic limit of a material frequently constitutes in practice but a very indifferent criterion of strength. By giving a cylinder an initial stretch before employing it in service its strength can be very greatly increased. This has long been known, and Professor Bridgman notes that the Austrians, so far back as the eighteenth century, attempted to increase the practical strength of their guns by forcing an expanding steel mandrel through the bore. In the United States Navy experiments, the stretching was effected by hydraulic pressure, the arrangement being as illustrated diagrammatically in Fig. 1. Here the gun is represented at G. The length of the bore under treatment is plugged by packing, the construction of which resembles that illustrated in Fig. 2, which will be described later on. A small hand pump H was used for filling the gun and raising the pressure up about 20,000 pounds per square inch, and this pressure was subsequently raised to over 90,000 pounds per square inch by means of an augmenter consisting of a cylinder C and a plunger P placed between the platens of a large testing machine. The fluid used for transmitting the pressure was kerosene. Other fluids are liable to give trouble by solidifying under extreme pressures. The intensity of the pressure was measured by the change in the resistance of a coil of manganese wire immersed in the fluid.

The plunger P of the augmenter was of glass hard tool steel, and the joint between it and the cylinder C was made tight by the packing illustrated in Fig. 2. This packing, it will be seen, consists of a plug K with a stem of smaller diameter. Around the stem are placed the packing rings. Of these those lettered A are of soft steel, whilst B and B denote rings of solder and C a ring of ordinary soft rubber, whilst D is a ring of hardened steel. The end of the stem of the plug K is cut off so as to leave a clear space between it and the end of the plunger P, thus providing for the compression of the packing as the pressure rises. The packing is, it is stated, tight under extreme pressures, but its friction is considerable. During the process of applying hydraulic pressure the increase in the outer diameter of the gun was measured at intervals and these strains were then plotted against the corresponding pressure as indicated in Fig. 3. On a first application of the pressure there was, as will be seen, no proportionality between stress and strain, but after repeated applications approximate proportionality is attained as will be seen from Fig. 4. Here the points observed during the release of the pressure lie approximately on a straight line.

The deviations from the straight line law are attributed to initial stresses in the forging which can only be eliminated by repeated applications of high pressures. After this hydraulic treatment the gun was finished and tested with proof charges, the results being highly satisfactory. Nine rounds were fired with charges developing a pressure of 21.8 tons per square inch, the service pressure being 13 tons per square inch. As already mentioned, the process is not new, and in our issue of March 12, 1915, page 306, Professor John Perry described a very similar series of experiments made under his direction first at Elswick and later elsewhere. Here, as in Professor Bridgman's experiments, the cylinder under treatment had a bore of 3 inches. Its external diameter was 10 inches and it was subjected to gradually increasing hydraulic pressures up to 40 tons per square inch and withstood subsequently the firing of a charge inside it developing a pressure of 52 tons per square inch.

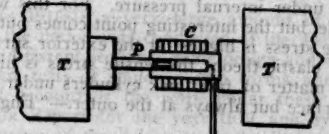
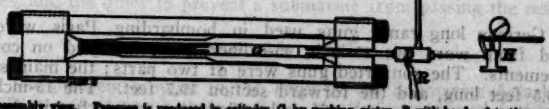


Fig.1.



General assembly view. Pressure is produced in cylinder C by packing piston P with heads of testing machine F. Pressure is transmitted through connecting pipe to interior of gun G. When R are connected to manganin resistance cage. Initial pressure is produced by hand pump H.

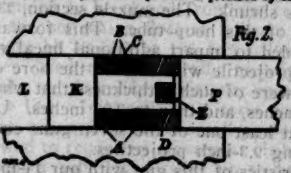
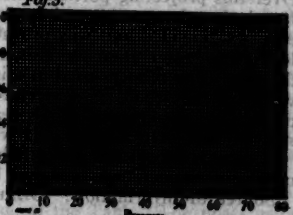


Fig.2.

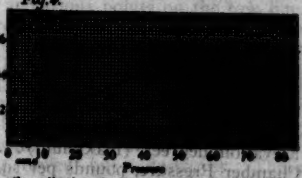
Details of packing plug of cylinder. Piston P pushes plug toward liquid at L. A is soft steel, B solder, C soft rubber, D hardened steel. E is unsupported area back of the plug.

Fig.3.



First application of pressure to the bomb. The increase of outside diameter (ordinate) is shown in inches against the internal pressure (abscissa) in thousands of pounds per square inch.

Fig.4.



Fourth application of pressure to the bomb. The increase of outside diameter (ordinate) is shown in inches against the internal pressure in thousands of pounds per square inch.

During the hydraulic treatment the cylinder was heated by gas jets to such a temperature as to prevent the metal hardening under plastic flow. Moreover, Professor Perry gave a theory of the resistance of such cylinders by which he claims that it is possible to calculate beforehand their strength under internal pressure. For this we must refer our readers to his article, but the interesting point comes out that according to his theory its tensile stress is highest at the exterior surface of the cylinder. On the ordinary elastic theory the tensile stress is highest at the internal surface, but as a matter of fact thick cylinders under pressure never do fail at the inner surface but always at the outer.—"Engineering," May 28, 1920.

CHARACTERISTICS OF THE GERMAN LONG RANGE GUN.

The German long range guns used in bombarding Paris were constructed from worn out 15-inch, 45-caliber guns, mounted on concrete emplacements. The converted guns were of two parts; the main section was 98.5 feet long, and the forward section 19.7 feet. The 15-inch gun was bored out, and a heavy uniformly-rifled tube 8.26 inches in diameter inserted, 42.3 feet of this tube projecting beyond the muzzle of the gun. Over this projecting portion a hoop was shrunk. The muzzle section, 23.6 feet in length was screwed to the end of this hoop tube. This forward tube was unrifled and was probably added to impart additional lineal velocity and better align the axis of the projectile with that of the bore of the gun. The reliners used in this gun were of such a thickness that when worn out they could be rebored to 9.3 inches, and then to 9.93 inches. Up to the time the firing on Paris ceased, at least one of the seven guns constructed had been rebored and was firing 9.3-inch projectiles.

A comparison of the general characteristics of this gun with our 14-inch naval gun, model 1920, is shown below.

Source of information: Ordnance Department.

COMPARISON OF GERMAN LONG RANGE GUN AND AMERICAN 14-INCH NAVY GUN.

	German 8.26-inch	American 14-inch Navy
Muzzle Energy, foot tons	467,250	85,750
Range, yards	132,000	39,000
Length, feet	118.2	59.7
Weight, pounds	318,000	180,395
Muzzle Velocity, feet per second	4,760	2,800
Elevation degrees (maximum)	55	38
Chamber Pressure, pounds per sq. in	44,000	40,000
Length of Projectile, feet	3.44	5.09
Accuracy Life, rounds	50	250
Weight of Projectile, pounds	264	1,400

—"The Military Engineer," March-April, 1921.

MODEL EXPERIMENTS IN CONNECTION WITH SUBMARINE WARFARE.*

By G. S. Baker, O. B. E.

§ 1. During the four years 1915-18 a very large amount of research work of vital importance to the nation was conducted at the Tank. This work can be divided into five different sections—mine design and sweeping, torpedo firing, seaplane design, tests of standard ship forms, and submarine catching. A very brief account of the work under these different headings is given in the Tank Report for the year 1919, and by permission of the Lords Commissioners of the Admiralty a detailed account of one section of the work is now placed on record. This section deals with the efforts made to cope with the submarine menace by means of nets hung in such channels as were used by the enemy submarines. These nets were of two kinds, one intended merely to indicate the presence of submarines, and the other to prevent a submarine from passing the net.

INDICATOR NETS.

§ 2. The problem brought to the Tank by the Submarine Attack Committee in connection with these nets was the design of a satisfactory float. The nets were light in construction, hung in strong steel wire frames, and intended to be ripped from the frame by a passing submarine. Attached to the net was a float to indicate the movement of the submarine which might be 50 feet to 100 feet below the surface. These floats, which had been tried mere generally unsatisfactory, either because of their instability, or their high resistance in any sea, which caused them to break the tow line. In April, 1915, tests were commenced in the Tank. The float was required to be as small as possible, make sufficient disturbance when towed to render its movement visible, to be directionally stable when towed from a submerged point up to at least 10 knots' speed, and to carry a weight of 23 pounds when at rest.

A number of floats were tested, all of full size, some being supplied by the Admiralty and five designed and made at the Tank. The floats were towed down the waterway, from the lower end of a strut which projected well below the water surface. As the tow wire in the experiments was very short compared with that used at sea, its resistance was augmented by looping on five lengths of wire rope at various points on the tow line. These tests served to show what was required, and the final Tank form was given a spoon bow and V-shaped transom. All the buoyancy was placed forward, and the after end was left open and so arranged that it would take all the necessary mechanism. The placing of the weight well aft and the V-shaped stern gave it the necessary stability at all speeds.

§ 3. These tests were followed by trials at sea, made on May 17 and 28, June 11, and July 2, 1915. The floats were towed from a "kite" 70 feet below the surface, the kite itself being towed by a destroyer. These showed that in fine weather a toboggan float (*i.e.*, one with a flat bottom intended to plane on the water) with a rear fin, could be made quite stable, but gave little or no splash. In rough water they were liable in short seas to receive a blow square to the flat bottom and part the towing wire—all the floats of this type tested, behaved in this way. The Tank form towed well in both rough and smooth water at all speeds up to 14 knots, and gave a good splash at all moderate speeds.

* Paper read before the Institution of Naval Architects, March 24, 1920.

BOOM DEFENSE NETS.

§ 4. The object of the experiments was to study the behavior of a defence net when attacked by a submarine, to determine the best method of supporting such a net and of holding the ends of the supporting jackstay, so that the submarine should be stopped without serious damage to the net. The experiments also served to show the most likely means by which any such defence net may be pierced and the subsequent motion of the submarine after impact. The experiments were made during April, May and June, 1916, to the order of the Booms Trial Committee.

§ 5. *General Considerations.*—The advance of a submarine in any direction may be arrested by a net in one of three ways:—

(1) By retarding its movement without slewing it.

(2) By deliberately converting its translatory motion into rotary motion—upwards, downwards or sideways.

(3) By loading the submarine with such weights that it must go to the bottom.

These experiments deal only with methods (1) and (2).

Any of these ends has to be obtained by means which give security against the attack of such craft on the surface, and the means provided must withstand the thrust of the tide and heavy weather.

The nets must be supported on floats, and to withstand the tidal thrust and surface attacks these floats must also be connected by strong wires or jackstays, secured at or loaded at their ends. To keep the nets from being swept to the surface they should be slightly loaded. If the submarine is to be brought to the surface, the net must be strongly secured to the jackstays. If it is to be thrown to the bottom, it must be secured to the jackstay, preferably by parting strips, and strongly anchored at several points along its length. With a loose bottom and very deep water with not much tidal movement, the latter method would result in the submarine meeting the bottom with heavy trim by the bow, and it would probably remain embedded in the bottom. This method of defense is dealt with in § 21. For shallow channels, method (1) is the only practicable one. For deeper and more important waterways the net must be able to work by either method (1) or (2), according to how it is attacked. In both these cases the retarding force is brought to bear upon the submarine, through the net—by its weight and the force in the wires. This force in the wires is transmitted to the jackstays and is balanced by the force at their ends and the water resistance of the floats. The net experiments, therefore, were arranged to determine the strength required in the net wires, when the net was attacked in various ways, and the effect of holding the jackstays in a rigid, semi-rigid or entirely free manner.

§ 6. For the sake of low resistance the nets are of large mesh. So far as weight is concerned there is little to choose between a mesh with vertical and horizontal sides, and one with its diagonals vertical and horizontal. With the former net, the stem of the submarine must always meet one or two horizontal wires which will "belly out" under the impact and be supported mainly by the vertical wires to the jackstay. The number of vertical wires taking the load will depend upon the depth at which the net is attacked, but will be at least two. With the diagonal mesh, unless the length of the diagonal of the mesh is made less than the average depth of the stem of a submarine (which means a smaller mesh than in the other), there is some chance of the stem entering the mesh. This contingency must, however, be allowed for in any case, as it is a comparatively simple matter to arrange the submarine stem so that it will open out the mesh and evade the wires. In both cases, therefore, there is a

strong chance of the submarine being held in the mesh and not by an individual wire. The horizontal wires, however, have a slight advantage over the diagonals in the fact that they are more likely to catch deck fittings, periscope, housings, etc., than the diagonals, as the apex of the diagonal may be subjected to an upward or lifting force sufficient to keep it clear of these things at the center. Both types of net have to cope with any horizontal cutters fitted on the fore side of the bow planes. With the horizontal and vertical mesh, the horizontal wires struck by the submarine (or one of the verticals on either side of it) must be broken before these cutters can act on the net, and with this broken wire the net will not hold the submarine again until it has passed the cutters. If the attacking vessel has a stem fitting, to ensure the stem entering the mesh, the stem will enter to the same extent in each case before the horizontal cutters are strained against the wires, but the vertical wire is strained through a bigger angle over the cutter and is therefore more liable to part. There is, however, less strain on the clips with the vertical wires, and on the whole there appears to be little to choose between the two types of mesh, and only the vertical and horizontal mesh—which appears to be more common in practice—was used in the experiments.

APPARATUS AND MODE OF EXPERIMENT.

§ 7. The waterway in which the experiments were made is 30 feet across, 12.5 feet deep and 550 feet in length. A defence net was hung across this waterway and a model of a submarine was allowed to impinge upon it in various ways at different depths and at various speeds. According to what feature it was required to test, the net, floats, jackstay or holding arrangement of jackstay were varied. The model used in the experiments represented in its mode of propulsion, its stability and its rolling (transverse and longitudinal) a modified "E" Class submarine. It had a displacement of 418 pounds and was 12.5 feet in length, representing on a scale of one-eighteenth full size, a vessel of dimensions:—

Length over all	225 feet
Beam over saddle bags	23.5 feet
Displacement, submerged	1,120 tons

On the same scale the nets used had a span of either 432 feet or 450 feet, a depth of 120 feet or less, a mesh either 12 feet square or 15 feet horizontal and 12 feet vertical.

In each experiment the steady speed of the model, the load on the jackstay, the rate at which the jackstay paid out, the damage to the net, and the behavior of the model were noted. The total number of experimental runs made against the nets is approximately 200, and no conclusion was formed unless it was found to be correct by one or more repeated trials.

§ 8. *The Model.*—This was built up of yellow pine planks and carefully shaped to lines provided by the Admiralty. The thickness of the shell varied from 1 inch to 1½ inches, but the superstructure was solid. There were three hatchways to give access to various parts of the model. The model was propelled by twin screws in the same position and of about the same diameter as those in the ship. These were driven by two electric motors fitted in the after part of the model, and obtaining their power from 28 Edison cells amidships. Fig. A shows the general arrangement. The propeller thrust bearings were fitted at the after end of the shafts. The couplings between the motors and shafts were loose, to allow for any want of alignment between the armature and the shaft, or

for any slight warping of the model. The number of cells used could be varied from 14 to 28 in steps of two, the regulating switch being controlled from outside the model. Fully charged, the model had a submerged speed corresponding to 11.7 knots for ship. The minimum speed was found by experiment to be 4.7 knots for ship.

At the after end the model was fitted with strong vertical and horizontal metal fins, exactly the same in outline as the fins and rudders on the ship, but strongly secured to the model. No guards were fitted around the propellers. Bow planes were fitted, but these were not movable, being secured to the model so that their plane was parallel to the deck, or top of superstructure. A conning-tower carrying two periscope housings was fitted, and jumping wires were run from the latter, to the stem, aft to a point on the superstructure at about one-third of the length from the stern. Two jumping wires were also fitted from the forward outer corner of each plane to the stem—one horizontal, the other slanting down to the heel of the stem. The stem was covered with thin sheet brass to prevent the net wires cutting the wood.

The vertical fin at the after end was adjusted so that when the model was quite free, submerged, and propelling itself, it would follow a straight course for some distance. To bias it so that it would turn to port or starboard a small fin, 1 square inch in area, was fitted about the superstructure aft, and could be fixed in any angular position. Holding irons were fitted to the model so that it could be handled from the sides of the Tank. These were placed only in such position as made it impossible for the net wires to foul them. The complete model was given several coats of white lead paint and final coats of shellac varnish, each end being afterwards painted white to render its movements visible when submerged. The complete model is shown in Fig. B.

The model was ballasted so that its transverse metacentric height when submerged was to scale 9.2 inches for ship. Its transverse rolling period (from port to starboard and return) was equivalent to 2.01 seconds for ship. The longitudinal radius of gyration of the model, found by oscillating the model in air when suspended bifilarly, was one-third the half-length of vessel. This was somewhat less than was desirable,* but it was not possible to place the ballast nearer the ends of the model without rendering the internal mechanism inaccessible.

§ 9. *The Nets.*—The material of which the nets could be made was required to be fairly hard, comparatively weak for its diameter, capable of being held together at the corners of the mesh, and of maintaining its shape and size in the water. Eventually, after trials of various materials, a wire of suitable gauge was adopted. The net was brought to its correct weight in the later experiments by suspending weights at each corner of the mesh, in some of the earlier experiments by weights at the bottom of each vertical, and in the first trials was left uncorrected.

It was important that there should be no slipping of the wires at the corners of the mesh, and, to ensure this, single reef knots were formed in both horizontal and vertical wires at every corner. This caused some loss of strength in the wire, and the break usually occurred at the corner. The actual strength of the net was obtained after the experiments with it were complete, by breaking out three horizontal and three vertical wires and testing them. The figure taken as the breaking strength of any wire in any case, was the mean of these three tests. The variation in the results of these tensile tests was about 9 per cent total between different nets and

* The longitudinal radius of gyration of the ship assumed to be a solid of uniform density throughout its mass, is 0.42 of the half-length instead of 0.33 as above.

28 Cells in Series. Twin Motors. Propellers.

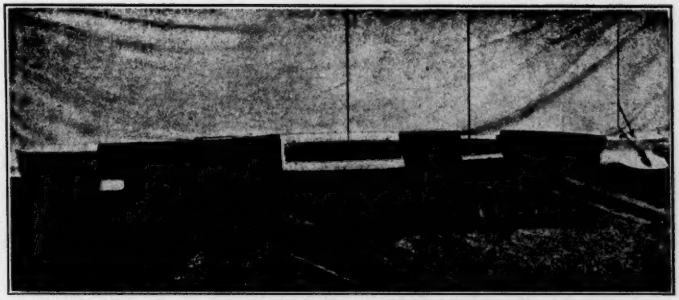


FIG. A. GENERAL ARRANGEMENT OF MODEL (MIDSHIP AND AFTER HATCHES REMOVED).

Bow Plane. Jumping Wire. Battery Switch. Bias Fin.

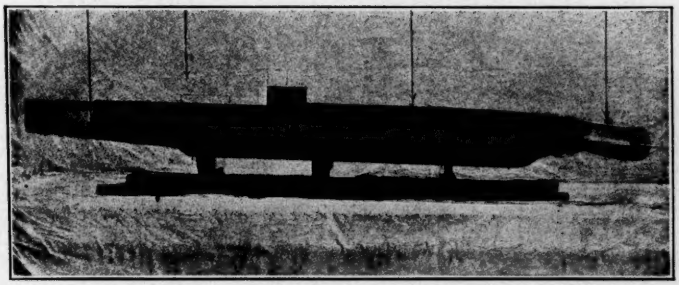


FIG. B. MODEL READY FOR EXPERIMENTS.

the wires in the same net. The wire used was ordinary trade wire and may have varied a little in diameter, but more probably the knot was made better is some cases than in others. Sinkers of the correct weight (to scale) were fitted at the bottoms of the verticals.

§ 10. *The Net Floats.*—These were made of solid pine when the originals they represented were of wood, and of thin brass plating when the originals were made of steel plating. The wooden ones were allowed to soak in the water and were then loaded down to an approximately correct water-line, when about one-quarter their volume was out of water. The models of the C and F floats (steel) were of about right weight as made.

The jackstays in the first experiments were made in short lengths fastened to eyes in the floats. This was found to weaken the jackstay too much, and in the later tests the jackstays were continuous across the whole section of net, with one turn on each eye in each float. The eyes holding these jackstays to the floats were in exactly the same position as in the full-size floats.

The upper ends of the vertical wires of the net were secured on the jackstays by knotting the wire over them, and were held from slipping by covering the knot and a short length of the jackstay with pitch. This hardened in the water and held quite well, and no slipping of this kind occurred in the experiments.

§ 11. *The Jackstays.*—These varied in number and strength with the type of floats tested. Where more than one jackstay were used, they were drawn together at each end about 1 foot beyond the last float, and secured to a strong string, by means of which it was kept in tension.

§ 12. *Tension in the Jackstay and Recording Gear.*—The apparatus used in the majority of the experiments is shown in Fig. 1. A weight was suspended in a pipe on each side of the Tank, and the suspending string, after passing over a pulley, was wound round and secured at its end to the 2-inch wheels at U. A second wheel, 4 inches in diameter, was secured on the same spindle and the string attached to the jackstay was wound round and secured to this pulley. The tension in the jackstays at their ends was therefore one-half the tension in the strings supporting the weights. The actual tension in the weight string varied during each experiment according to the rate at which the jackstay was moving, and arrangements were made to measure the first part of this movement when the acceleration was likely to be appreciable.

A light carriage working on levelled rails was attached to the weight string (see Fig. 1). This carriage carried on its upper surface a sheet of cardboard. A dotter making 50 vibrations per second was arranged so that it just touched the cardboard when at the maximum amplitude of its oscillation. As the jackstay moved out and the weight lifted, the dotter recorded the distance traveled at every 1/50 second. This dotter was in the same electric circuit as a tuning fork with which it had been synchronized, and could be kept vibrating for any length of time.

§ 13. *Speed and Direction of Model.*—There were two possibilities of propelling and directing the model:—

(1) To have it free throughout the experiment, controlling its motion by setting its fins and rudders before starting it.

(2) To hold it by the carriage which travels over the waterway, giving it bias in any direction, and releasing it some 10 feet before reaching the net—the propellers working at the necessary rate for propulsion at this speed.

Both of these methods were tried. For (1) the model was started from a punt 70 feet from the net, and the time taken to travel the last 35 feet just before striking the net was noted. This start was necessary to allow

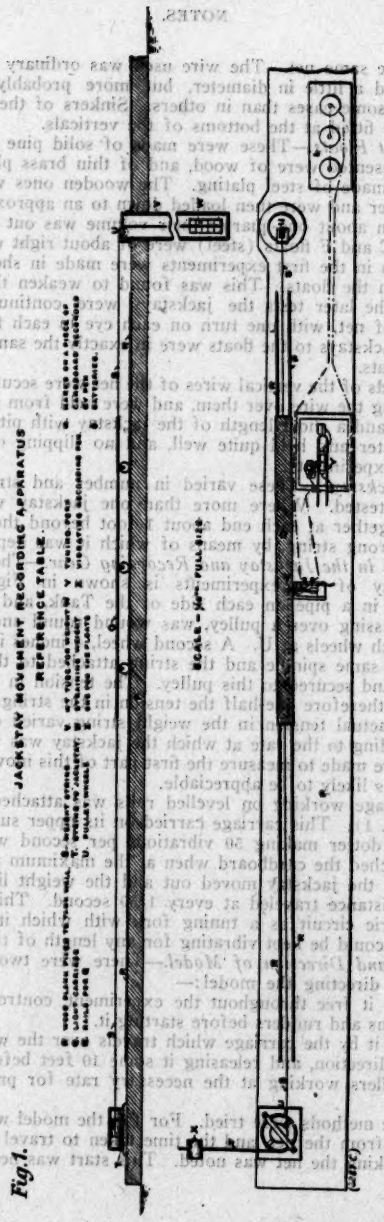

JACKSTAY MOVEMENT RECORDING APPARATUS
REFERENCE TABLE
SCALE - ½" = FULL SIZE

Fig. 1.

the model to attain a reasonably uniform velocity, and a greater length would have been better, but even at this distance the point of impact was very uncertain, and only the first tests on the B net were made in this way.

In method (2) the model was held to the traveling carriage by two vertical struts, one gripping the model by a short piece of bar between the periscope housings, the other holding the horizontal stern fin in such a way as to allow the model to run out forward when the carriage stopped. The propellers were set in motion with the desired number of cells in circuit, the carriage moved towards the net at a predetermined speed, and the model was released from the forward grip when the stem was about 20 feet from the net. The number of cells necessary for propulsion was determined by the earlier tests already mentioned and occasionally checked by running the model (free) under the traveling carriage. The model could be directed to any portion of the net by starting it with bias to port or starboard, or by immersing it to different depths on its struts. This method was adopted in all but the first two days' tests.

§ 14. *Experiments.*—These took three different directions, the study of (1) individual boom defence nets as fitted or proposed, (2) the methods of holding the jackstay and supporting the nets, so that the defence remained intact when attacked, (3) the determination of the necessary strength of the net wires and the general behavior of net and submarine.

In each case the net was attacked on the surface at speeds possible with submarines when running on the surface; and at various depths below the surface, and in some cases the attack was made nearer one support than the other. In the general study of submarine, net and jackstay movement, a cinematograph was used and photographs were taken to illustrate the general behavior.

The following may be taken as fairly descriptive of what happened when a net was attacked by a submerged submarine, the net jackstay rendering when the load reaches a certain amount and maintaining a constant tension as it renders.

When the submarine struck the net, the first movement of the latter was to form a local bulge with the stem of the submarine at its center. The thrust of the submarine was transferred to the jackstay by the wires at and near the bulge, and jackstay and floats would move in the direction of advance of the submarine. As the jackstay bent, the whole strain in it was transferred to a few "verticals" (mainly) at the bulge, which also had to take the resistance of net and floats as these were dragged through the water. At low speeds and with perpendicular impact, no part of the submarine except the stem and its jumping wires touched the net, and the system would come to rest with the net on the bow of the submarine, and the thrust of the screws balanced by the drag of the wires. With impact at or about 24 feet below the water surface, the same remark held good, unless the jackstay tension was very light, and not sufficient to balance the screw thrust. With heavy pulls in the jackstay the impact which was necessary to form a deep bulge was sometimes sufficient to break the net wires before the bulge could form. There was still the possibility in this case of the net holding the model if only a single wire had broken, by a wire fouling the fore end of the ballast keel, or by the double mesh jambing round the vessel, the submarine coming to the surface under the pull of the net, and the jackstay catching the top of the periscope housings, if the jumping wire was not taut.

§ 15. When the net was struck near the bottom, a horizontal wire usually held on the stem (or the stem entered a mesh). The angle of the bulge, which the stem formed in the net, would get smaller the farther the net advanced under the impulse of the submarine. But as the net

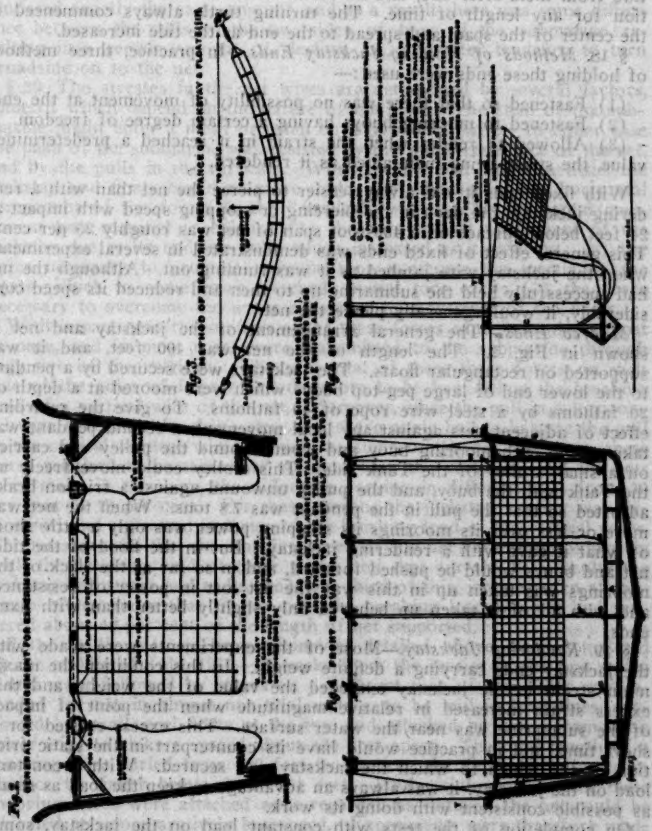
was pushed out, the submarine was being pulled backwards and upwards by the "vertical" wires, and this upward pull in the net not only lifted the submarine to the surface, but being exerted at the extreme end of the ship trimmed it heavily by the stern. As the submarine lifted, the net settled more on the bow and there was less chance of it being pierced. Moreover, when the vessel had stopped advancing, the jackstay was still pulling it backwards, and in some cases the trim of the model increased very considerably during this backward movement.

If the model impigned on the net out of the perpendicular, the net action was very similar. The bulge in the net was necessarily not symmetrical, and there was a strong tendency as the submarine stopped advancing for it to slew round and foul the net with its propellers.

§ 16. Although the forms were originally intended for defence against submerged craft, the jackstay connecting up the floats and supporting the nets was made very strong to support attack from surface craft. A number of sigh speed runs (14 knots to 18.8 knots) were therefore made with the conning-tower and stem partially out of water. A load on the jackstay end, practically double that required for attack submerged, was necessary if the model was to be stopped with the movement possible with the Tank apparatus. The jackstay formed of three wires in a horizontal plane was not so satisfactory as one with two wires, one above the other—in the former the strain was sometimes taken on the center wire only (particularly if attacked with the stem just flush with the water when the bow wave tilted the floats up and threw the first wire clear of the stem), and this also had to support the net resistance and sometimes gave. These "surface" experiments, however, showed that the loading required was beyond what was practicable if the net was not to be pierced when attacked, so that the nets required very considerable strengthening to cope with all possible chances of submerged attack with such heavy jackstay loads. A far more satisfactory arrangement would have been a strong and heavily-loaded boom for surface attack quite separate from the nets and their jackstay, which could then have been arranged for submerged attack only.

The limitations of the stopping powers of the boom defence nets at a number of different ports were obtained in this way. These nets varied in depth from 24 feet to 120 feet, but the results with all of them were very consistent as regards speed limitations and the difference in result of attacking close to the surface or at great depths. The experiments led to the detection of more or less serious defects in some cases, and to the general strengthening of all defence nets, the originals being too weak for their purpose.

§ 17. The nets were invariably fitted in practice across tidal waters, and one branch of the work was to determine what was the effect of tidal wash upon the jackstay and floats. For this purpose the jackstays and floats were towed up the Tank. The jackstay was allowed to sag a definite amount by increasing the length to somewhat more than the span. A skeleton diagram of the arrangement is shown in Fig. 3. The total pulls in the jackstays to maintain definite and different sags was obtained in this way. These pulls were always much greater for the types with wood floats than for those with steel floats, the excess varying from 30 per cent to 80 per cent. This was partly due to the latter being smaller, and to their better shape and arrangement. This pull in a tidal stream was most serious in some cases, amounting in a 4-knot tide to several tons. It constituted a definite handicap on the types with wooden floats as the greater jackstay pull necessary with them lowered the "stopping speed" of the defence.



In most cases the jackstay consisted of several wires, and it was found that with any very definite sag, if these wires were in a horizontal plane the floats became unstable in tides of 3 knots to 4 knots, and turned turtle, tangling the whole system. This was due to the outer wire of the bend taking the major strain, and the floats were therefore held against the tide from their after end, and no rectilinear float will maintain this position for any length of time. The turning turtle always commenced in the center of the span, and spread to the end as the tide increased.

§ 18. *Methods of Holding Jackstay Ends.*—In practice, three methods of holding these ends were used:—

(1) Fastened so that there was no possibility of movement at the end.
(2) Fastened to mooring buoys having a certain degree of freedom.
(3) Allowed to render when the strain in it reached a predetermined value, the strain being maintained as it rendered.

With fixed ends it was always easier to pierce the net than with a rendering jackstay, the drop in the piercing or stopping speed with impact at 24 feet below surface on a 430-foot span of net was roughly 25 per cent. This general effect of fixed ends was demonstrated in several experiments when the jackstay wire jambed as it was running out. Although the net had successfully held the submarine up to then and reduced its speed considerably, it would generally pierce the net.

Moored Ends.—The general arrangement of the jackstay and net is shown in Fig. 2. The length of the net was 400 feet, and it was supported on rectangular floats. The jackstays were secured by a pendant to the lower end of large peg-top buoys which were moored at a depth of 20 fathoms by a steel wire rope of 40 fathoms. To give the retarding effect of adjacent nets against any local movement, a second pendant was taken from each mooring buoy and wound round the pulley and carried on a small trolley of the Tank side. This trolley could move freely up the Tank with the buoy, and the pulley unwound against a friction brake adjusted so that the pull in the pendant was 7.8 tons. When the net was more or less over its moorings its stopping power was only a little short of what it gave with a rendering jackstay. But in the flood of the tide, net and buoys would be pushed forward, and in so far as the slack of the moorings was taken up in this way the net lost in power of resistance, and with all of it taken up behaved only slightly better than with fixed ends.

§ 19. *Rendering Jackstay.*—Most of the experiments were made with the jackstay ends carrying a definite weight. In this condition the maximum strain in the jackstay exceeded the value of the weight, and this excess strain increased in relative magnitude when the point of impact of the submarine was near the water surface. This excess existed for a short time, and in practice would have its counterpart in the static friction of the winch to which the jackstay was secured. With a constant load on the jackstay, it was always an advantage to keep the load as small as possible consistent with doing its work.

On completion of the tests with constant load on the jackstay, some further experiments were made with the load increasing as the jackstay ran out, but starting with a small initial value. Unless the model after impact with the net was allowed to advance beyond what was practicable in the experiments, the initial load could not be reduced much, without the necessity of a very large (and therefore dangerous) final load.

When the forward movement of the submarine had been destroyed the backward pull on the jackstay was always greater than it was with the constant load. This pull as a rule was exerted at and about the bows

of the model and was sufficient to overcome the propeller thrust and push the model backward, *i.e.*, stern first for some distance. As it went astern, the trim it had received from the net as it was pulled up was greatly increased, and reached as high as 50 degrees in one or two cases, against about 30 degrees with a constant load on the jackstay. One other effect of this varying load on the jackstay was noticed. The two ends seldom rendered exactly the same amount, and as a result there was some difference between the pulls on the two sides of bow of the submarine, resulting in a certain amount of slewing, *i.e.*, a greater tendency to turn broadside on to the net.

§ 20. The stresses in the net wires are determined by several factors, some of which depend on the nature of the blow. The stem of the submarine might enter a mesh or catch on a horizontal wire. In either case the push of the submarine was resisted by the water resistance of the net, and by the pulls in the verticals. When the net had bulged a little, the jackstay and floats began to move under the influence of the horizontal component of the forces in the "vertical" wires. This horizontal component was required to balance (1) the fore-and-aft component of the jackstay due to its endwise pull—which increased as it paid out; (2) the force necessary to promote movement or overcome the inertia of the floats—which in some cases weighed several tons each; (3) the force necessary to overcome the water resistance to movement of the floats.

With impact near the surface, all of these forces came into play almost at once, and in most cases were transmitted to the submarine through only two of the verticals. The sum total of these three forces on each side of the point of impact had therefore to be kept below the breaking load of a vertical wire of the net. Moreover, since the pull in a horizontal wire bent over the ship's stem at a sharp angle had to balance the pull in the critical vertical wires, as well as some of the net resistance, these wires required to be somewhat stronger than the verticals. All the nets were made in this way, the difference varying in different nets from 30 per cent to 12 per cent. For these reasons the jackstay loading was determined and expressed in terms of the breaking load of the vertical wires. The tests showed that the stopping speed of any net varied approximately with the breaking load of these wires, and that for the best results the jackstay end strain had to be about one-third of this, increasing to a maximum of not more than 80 per cent the breaking load when it had rendered about 40 per cent of the length of net supported.

§ 21. *Paddle Nets.*—The experiments detailed above were completed by the middle of July, 1916—14 weeks from the receipt of the first order—and were followed by others on a different type of net. This was intended for use in almost non-tidal waters, to be laid in long lines, supported by small steel floats at the top, and attached by wires from their lower edge to sinkers spaced about 120 feet apart. The total length of a mooring wire varied from 100 feet to 200 feet, the depth of water from surface to sinker being about 174 feet. If the sinkers to which these mooring wires were attached were regarded as fixtures, the net could be defeated at a comparatively low speed. Experiments were therefore made by the Boom Trials Committee to determine the forces necessary, first to break out a sinker in different bottoms, and then to drag it over the bottom. In the later Tank experiments the lower ends of the moorings were allowed to move when these breaking-out loads had been reached, the load being arranged to drop quickly to that corresponding to a dragging sinker.

Only a comparatively short length of net could be used in the Tank, and it was necessary to represent in some way the effect of adjacent portions

of net, mooring wires and sinkers. Adjacent sinkers would strain principally along the bottom but the inertia and resistance of adjacent net would be felt on all the horizontal wires of the net.

The experimental apparatus as adopted is shown in Fig. 4. The net was of the same mesh as before, but the end verticals near the Tank walls were replaced by light flexible battens, and the horizontal wires were attached to rings which could slide up or down the batten. From the top end of the batten a string was taken to a pulley on the tank wall. A similar string was taken from the lower end of the batten to a pulley at the tank bottom, and thence to another on the roof. These strings were loaded in various ways to give the effect of the drag of adjacent nets, and the manner and time (relative to the movement of net) at which each came into play were also varied.

The pull in the mooring wires was regulated by taking the wires over pulleys on the Tank bottom and thence over pulleys on the roof. These wires were loaded each by two weights d and c, the latter close up to the roof pulley, and, attached to each wire on the "net" side of the pulleys a on the tank bottom, was a weight, h. Between h and the net the mooring wire was quite slack, but when the net had moved under the impact of the submarine and taken up all the slack of the mooring, if the tension reached the sum of weights c and d the sinker would commence to drag. When it had dragged one or two inches the weight c moved to the other side of the pulley and the tension in the mooring dropped to $(d - c)$. The weights $(d + c)$ and $(d - c)$ were arranged to give the breaking-out and dragging loads for the sinker used.

The diagram (Fig. 4) shows the movement of the net, the line of attack of submarine being represented by an arrow. The trim of the model produced by impact was never very great, the general result being to pull the model down more or less bodily. Occasionally the net would slew the model badly, owing to unequal movement of sinkers or to impact at an angle, and this slewing continued after advance had ceased, until the propellers fouled the net.

§ 22. With some adjustment of the relative strength of wires, this type of net was approximately equal in stopping power to that already described. If the sinkers in actual practice "grew into the bottom" they would act as fixed moorings and the stopping power would suffer. But the moorings could be arranged to give a quite good margin over the experimentally determined breaking out load.

§ 23. It will be easily understood that it is not possible to give the many detailed conclusions arrived at in the course of the experiments, as submarine attack on harbors, etc., is still to be reckoned with in the future. The experiments were witnessed on a number of occasions by various Admiralty and Port officials, and on June 14, 1916, a number of experiments were witnessed by the First Lord and Third Sea Lord of the Admiralty. The experiments were arranged in conjunction with Captain Learmouth, Captain Seymour and Commander Bircham, of the Boom Trials Committee. The details of the experiments were in part due to Commander Bircham who took charge of the cinematograph work, and Lieutenant Powell of the Cameronians. It was due to the energy and willingness of these and other members of the Tank staff, and particularly to the mechanics, that the first portion of this research work on defence nets was carried out in the short space of 14 weeks. The final report on the whole matter was sent off in 19 weeks from the receipt of the first order.—"Engineering," June 4, 1920.

RIVETS AND RIVET SPACING.

Some Notes on the Size of Rivets and Their Spacing for Oil-tight Work.*

Rules for rivets and rivet spacing are the result of evolution arrived at by rule of thumb, experience, size of rivet found practicable to drive, spacing suitable for caulking, experiments, calculations, survey reports on damage cases, etc., and are, in the opinion of the writer, generally satisfactory with very few exceptions, which will be referred to later. Oil tightness depends, perhaps, more on good workmanship than any other one thing, especially so far as riveting is concerned. If every rivet could be guaranteed 100 per cent perfect, slightly greater spacing would be practicable and might be allowed by classification societies except where a given efficiency of joint is essential. The same standard of workmanship does not exist throughout the yards of the world or in some cases even throughout the same vessel.

EARLY TANKERS.

While there were probably some barges in existence prior to 1872, it was in that year that the first bulk oil-carrying steamer was built at Palmers, Newcastle-on-Tyne, England. Although this vessel was built especially for carrying oil in bulk, she had a double bottom, and the indications are that the rivet spacing was the same as in vogue at that time for watertight work. Following this date, new bulk oil carriers were built from time to time, but perhaps for about three decades converted vessels contributed more or less to increasing the fleet. During this period the spacing of rivets, even for watertight work, was in some respects greater than at present and vessels frequently leaked a little when new, but they were built of iron and the joints soon rusted up tight, thereby increasing the friction of the riveted joints and resulting in stronger and more durable vessels under existing conditions. This rusting-up process, however, would not take place in the case of new vessels built to carry crude or lubricating oils in bulk, and on that account some of the converted vessels were tighter than new tankers. It was during this period that the importance of the friction referred to above became more generally appreciated.

Oxidation, however, takes place in tanks carrying petrol, and this rusting-up process, undoubtedly, penetrates into the seams. On this account it might be beneficial to carry a cargo of this character on the first voyage or two as is sometimes done. Tests have demonstrated that, while the friction of the joint has no effect on the ultimate strength, it has a safe factor above working stresses where workmanship is good. If there is the slightest slip, defective rivets will come to light; therefore, in oil-tight work intimate metal-to-metal contact is of the greatest importance for permanent oil tightness, especially where subject to alternating stresses set up by the working of the ship.

PROGRESSIVE RIVET FAILURE.

The early tankers, both new and converted, had relatively less beam and were consequently easier in a seaway, yet the penetrating character of the oil sought out weak spots and loosened rivets developed under new stresses created by carrying oil in bulk. The first leaks frequently oc-

* Abstract of Paper read before the American Society of Naval Architects and Marine Engineers by Mr. H. P. Frear.

curred at the ends of stiffeners and in rivets connecting frames to shell which were subject to stresses not common in ordinary vessels. After these started the vessel began to work more, and progressive damage followed, frequently resulting in almost entirely reriveting the vessel. The old reliable pan head was found to produce the most intimate contact and to be more permanently oil-tight, and was recommended except where some other types were absolutely necessary.

Up to 1894 there were built or building to Lloyd's class sixty-four tankers, yet the Society did not consider that it had sufficient experience to lay down definite rules for their construction. In this year, however, Mr. Benjamin Martell, Chief Surveyor to Lloyd's Register, read a paper before the Institution of Naval Architects wherein he outlined his views on the details of construction and of riveting for bulk oil carriers, recommending three diameters for oil-tight work. This paper in pamphlet form was distributed and for a number of years served as a guide to shipbuilders in the design and construction of tankers intended to be classed at Lloyd's. Within a comparatively short time after this, Bureau Veritas, apparently profiting by this paper, published rules for tankers, and the American Bureau of Shipping published their first rules in 1902, but the first rules of Lloyd's did not appear for general use until 1909. Both Bureau Veritas and the American Bureau recommended three and one-half diameters at the start, while Lloyd's specified one more rivet in a frame space than required for watertight spacing resulting approximately in three and one-half diameters.

Evolution was a slow process, and no doubt in numerous cases the true cause for failure was not recognized. Mr. Martell realized this when he wrote his paper and provided for tanks not over 24 feet in length, increased stiffeners and brackets, as well as closer spacing of rivets. Had the earlier tankers been better designed as regards tanks, stiffeners, brackets and compensation, where needed, the question of rivets and spacing of rivets for oil-tight work might possibly have received somewhat less attention during those troublous times. This appears to be a reasonable inference from the fact that so many converted tankers have since given long and satisfactory service. That oil-tight work should be double riveted was an accepted rule. This is still the general requirement, but single riveting is now permitted in the seams of double bottoms intended for carrying fuel oil.

OIL IN DOUBLE BOTTOMS.

This concession was not made until after many vessels in commission had been converted to carry fuel oil in their double bottoms with single-riveted seams and their class continued. Both new and old single-riveted tank tops, where fuel oil is carried in double bottoms, have proved a success. Why should there be any greater risk in single-riveting the seams for decks and trunks of tankers, which are subject to less test head and deflection, especially where the spacing is made closer for oil-tight work? Just how far this argument could be carried for tankers intended or liable to carry bulk petrol or other light distillates is an open question. If single-riveted seams are tight for crude oil, would they be as tight or rust up tighter with petrol? Lloyd's do not permit gasoline to be carried in single-riveted double bottoms, but it is not known whether or not it would be permitted even with double-riveted seams with dry cargo above.

Lloyd's formerly recommended single double-riveted bulkhead bounding bars, but later gave the builder the option of using double single-riveted bars. The general practice now is to use double single-riveted bars

with countersunk heads and points and bars caulked on both flanges on each side. Classification societies now allow a spacing of five diameters connecting bulkhead bounding bars to shell. The writer is inclined to favor single double-riveted bounding bars and pan-head rivets in standing flange with caulking on one side, because it is believed this lends itself to cheaper and better workmanship and would insure the parts being drawn up more closely. Countersunk heads are, of course, unavoidable in the shell flange. On the Clyde, bulkhead bounding bars are frequently hydraulically riveted on the ground, using pan-head rivets and finishing with snap heads and points.

USE OF TEE BARS.

Where double bounding bars are fitted, careful workmanship is necessary to insure the two shell flanges lining up perfectly for intimate contact. To overcome this difficulty and with a view to better and more economical work, T-bars have in some cases been used, although the present available 6½-inch by 6½ inch T-section does not provide sufficient width of shell flange on that side on which the bulkhead plating connects to the standing flange for best spacing of the rivets. A more general use of this type of bounding bars has been recommended by some owners, and with this in view two of the large shipbuilding concerns have without success urged the steel companies to roll a larger section. Conditions, however, would be substantially the same as with single bounding angles. Pan-head rivets could be used except in one flange, three-ply work would be eliminated, and only two edges would require caulking. A single angle as recommended above would have the same number of rivets, the same amount of caulking, and be more easily set. In gunwale bars, Lloyd's require four and one-half diameters for watertight work and five diameters for oiltight work, which does not seem consistent. The table on the next page gives, for a number of selected parts, the rivet spacing recommended by Mr. Martell in 1894 for oiltight work and present requirements for both oil and watertight work of Lloyd's, American Bureau, United States Navy, and British Admiralty.

The extensive use of oil fuel in naval vessels during the last few years has brought up for consideration and settlement many problems in design which did not exist in coal-burning vessels. In vessels of great length, such as scout cruisers and battle cruisers, longitudinal bulkheads are used as strength members. The riveting connections in these bulkheads must be designed to obtain maximum strength and, therefore, the riveting should not be so closely spaced that more material is punched out of the plating than allowed in the calculations, without compensation. If, however, these bulkheads form boundaries of oil tanks, it becomes a matter of importance, when transverse bulkheads join the longitudinal strength bulkheads, to avoid too close spacing in the bounding bars. If the strength bulkhead is designed for a spacing of six diameters for rivets in stiffeners and oil tightness demands five diameters at a transverse bulkhead, it becomes necessary to introduce compensating liners which add to the weight and, on account of the extra thickness of plating, increase the difficulty of making the structure tight.

TESTS FOR OIL-TIGHTNESS.

The Hull Division of the Philadelphia Navy Yard recently made a test on an experimental box to determine if a six-diameter spacing of rivets would be satisfactory under the conditions as stated above. Six diameters might be quite suitable on a deck near the neutral axis, but unsuit-

	Martell, 1894.	Lloyd's 1919-20 rules.		American Bureau 1919-20 rules.		U. S. Navy		British Admiralty	
	O.T.	O.T.	W.T.	O.T.	W.T.	O.T.	W.T.	O.T.	W.T.
Shell plating seams . .	3	3½	4	3½	4½	3½	3½–5	4	4–4½
Deck plating seams . .	3	3½	4½	3½	4½	3½	3½–5	4	4–4½
Bulkhead plating seams	3	3½	4½	3½	4½	3½	3½–5	4	4–4½
D. B. plating seams . .	3	3½	4	3½	4½	3½	3½–5	4	4–4½
Tunnel plating	3	3½	4½	3½	4½	—	—	4	4–4½
Frames to shell . .	6	6	7	6	*	8	—	4	7–7½
Deck bars	—	5	4½	—	—	—	4½	4	4½
Bulkhead bounding bars	3	5	5	5	5	5	5½	4	4½
Bulkhead stiffeners . .	6	6	7	†	†	6	6	6	7
Tunnel stiffeners . .	6	6	7	7	7	—	—	6	8
Shell butts	3	3½	3½	3½	4½	3½	4½	4½	4½–5
Deck butts	3	3½	4	3½	4½	3½	4½	4½	4½–5
Bulkhead butts	3	3½	4½	3½	4½	3½	4½	4	4½ 5
D. B. butts	3	3½	4	3½	4½	3½	4½	4	4½

* 7 up to 27-in frame spacing.
6 above 27-in frame spacing.
5½ in. deep tanks.

† 7 generally.
5½ in. deep tanks.
4 for 15 per cent length each end.

able for parts subject to stresses resulting from the vessel working in a seaway or from shocks of gun-fire or explosion. In this test it was found that the caulking of bounding bars to heavy bulkheads was satisfactory under a 60-foot head with a six-diameter spacing in the connection of transverse to longitudinal bulkhead and five-diameter spacing in bar connecting to the shell. No gunning was used and no special effort made to have the box absolutely tight on the first test. A similar tank was also recently tested at the New York Navy Yard to demonstrate the efficiency of six diameter spacing for T-bar stiffeners and for bounding bars to bulkheads. The tank was first subjected to a test head of 65 feet of water and found to be tight; then to a twenty-four hour endurance test with oil at the same pressure. Then the pressure head was increased to 93 feet, which pressure was maintained for four hours without leaks. The results of the tests appeared to indicate that in fairly heavy plating, where the members are not under the influence of other stresses than those due to head, six diameters are suitable. The thickness of angle bars, it was suggested, should preferably be not less than ⅜-inch. It is understood that further tests were directed to be made on experimental tanks at navy yards, but, unfortunately, full particulars are not at hand.

It has been the practice in naval work to use a spacing of three and one-half diameters for oiltight work in seams and straps, four diameters in angles to plates including staples, and four and one-half diameters on bulkhead stiffeners if caulked oiltight. This is a closer spacing than is required for watertight work. Experience in the construction of oil tanks in double bottoms seems to indicate that, with ordinary good workmanship, watertight spacing will be satisfactory for oil. While the naval authorities are not prepared to make the spacings the same, a compromise has been made allowing a somewhat greater spacing than called for by the navy rules for oiltight work. For single riveting three and one-half diameters are retained, but four and four and one-half diameters, depending on the thickness of the plates, are now accepted; five diameters are used in bounding bars of oiltight floors.

SUGGESTED RIVET SPACING.

One of the larger builders, as a result of mathematical analysis, has suggested the following spacing as permissible, provided that considerations other than oil-tightness did not determine the spacing:—

Diameter of rivet.	Thickness of plating.	Spacing in diameters.
¾	5/16	3.6
¾	⅜	4.2
⅞	½	4.5
1	¾	5.6
1⅛	1	6.3
1¼	1¼	7.0

The above spacings were based on the supposition that three and one-half inch diameters were satisfactory for a 10-pound plate using ⅝-inch rivets, and assuming that the deflection of the plate edge caused by caulking probably varies as the ratio of the fourth power of the distance between rivets to the fourth power of the thickness of the plating, and that the extension of the rivet varies as the ratio of the distance between rivets to the area of the rivet. In order to make practical use of the above it was suggested that watertight spacing be used for rivets in oiltight work on naval vessels.

A naval constructor stationed at the same yard carried the investigation further by considering the deflection that is caused by caulking and by the amount of general deflection of plating due to water pressure, and suggested that, based on the assumptions made, the following spacing would be satisfactory:—15-pound plate, 4 diameters; 20-pound plate, 4.5 diameters; 25-pound plate, 5 diameters; 30-pound plate, 5.3 diameters; 35-pound plate, 5.6 diameters.

The above tables were deduced without reference to strength of joint, and it was pointed out that this consideration might be more important than questions of tightness. For ⅝-inch rivets in 10-pound plating it was shown that approximately .006 inch is the deflection or amount of seam that can be caulked off without exceeding the elastic limit of the material. The stress on the rivet for such caulking was found to be very moderate.

THEORETICAL CONSIDERATIONS.

The analysis used in deriving the above table of rivet spacing for oiltight work was made to confirm a theory that every unnecessary rivet is a possible source of leakage, and that for heavy plating a wider spacing would be permissible, but considered only the stresses on the plating and

rivets due to caulking to the exclusion of the more complex stresses experienced under service conditions.

For thin plating, where large rivets in relation to thickness are used, the diameter is based more nearly upon theoretical considerations, but if the same considerations were followed in determining the diameters for thicker plates, the rivets would exceed the limit practicable to drive, and spacing based even on using the same number of diameters would at some point become greater than indicated in the tables. Thick plates, however, are used where greater strength is required, and as the rivets used are much smaller in relation to thickness a greater number is necessary to develop the strength of the plating. The uniform spacing in diameters adopted in practice, substantially, for all thicknesses of plating is a great convenience, and it is only necessary to increase the number of rows of rivets where greater efficiency of joint is desired. If the spacing suggested in the above tables is suitable for oiltight work, it might be necessary to increase the number of rows to insure proper strength and friction of the joint, in which event there would be an increase in weight and the attempt to reduce the number of rivets defeated.—"Shipbuilding and Shipping Record," Jan. 6, 1921.

BILGE KEELS OR GYRO-STABILIZERS.

The stabilization of ships is a problem which has always held a certain amount of fascination for the naval architect, and many methods have been brought forward to minimize the rolling of a ship which occurs when she is subjected to the action of the waves. The simplest form of stabilizer is the bilge keel which, while it adds a small amount to the cost of construction and to the resistance of propulsion, costs nothing for upkeep, and does not in any way detract from the carrying capacity of the hull. The anti-rolling tank has the advantage that it does not need machinery to operate it, the action of the water which tends to prevent the vessel from rolling being called into play by the rolling of the vessel itself; but these tanks occupy a considerable space in the hull which might be far more profitably employed in the carriage of cargo. The gyroscope represents a far more exact method of damping out the vibrations of a ship, but in addition to the cost and the space occupied, a certain amount of power is required to keep the gyroscopes in motion. For first-class passenger liners and for war vessels some means of preventing rolling is, however, very desirable, particularly in the case of war vessels from which accurate gun-sighting has to be performed, even when the vessel is exposed to the disturbing action of rough seas, and clearly some method of determining the actual stabilizing action of any proposed device should prove of great value to the naval architect.

The equipment of ship model tanks with wave-making apparatus has led to the performance of many experiments having for their object the determination of the exact behavior of a ship when exposed to the action of a seaway, but it has been left to Commander William McEntee, of the United States Navy—whose work in model experiments is well known and highly valued in this country—to employ the tank for comparing the stabilizing effects of the bilge keel and the gyroscope on models which have been subjected to rolling. In a paper recently contributed to the American Society of Naval Architects* he has described a series of ex-

* Comparative Tests of Bilge Keels and a Gyro-Stabilizer on a Model of the U. S. aircraft carrier *Langley*, by Commander W. McEntee.

periments carried out on the model of the United States aircraft carrier *Langley*, having this end in view. The *Langley*, a vessel of 520 feet long and 11,000 tons displacement, is being fitted for the carrying of aeroplanes, which are both to fly from and to land on her deck and it is apparent that the prevention of excessive rolling is a matter of even greater importance with vessels of this class than with other types of warship. A model to one twenty-sixth scale was employed and fitted with bilge keels, rolling experiments being made in still water, both with and without the keels, in order to determine the quenching power of the keels. It may be of interest to quote the results of this experiment. Without bilge keels the angle of rolling was reduced from 11 degrees to 3 degrees in about 54 swings, as compared with only eight swings when the keels were fitted. In making the tests, a gyroscopic roll recorder was employed, so that a continuous record was obtained showing how the rolling died out, typical curves being published with the paper.

For the comparative tests with the gyro-stabilizer, the model was rolled in waves of various heights produced in the model basin, these being equivalent to heights between 11.9 feet and 19.5 feet, as compared with the ship, and the period of the wave being so controlled that it produced marked rolling of the model. The gyroscope was operated by an electric motor producing a stabilizing moment of about 20 pounds-foot on the model, and the angle of roll was quickly reduced from 34 degrees to 3 degrees when the stabilizer was cut in, while with the same system of waves the bilge keels only reduced the angle of roll from 26.4 degrees to 9.6 degrees. Here again the diagrams of the vibration of the model show very eloquently the action of the gyroscope in damping out the vibrations as compared with the bilge keels. Other experiments were made in which very heavy rolling of the model was produced, and with an angle of roll of 48 degrees the stabilizer, when cut in, almost instantly reduced this to about 2 degrees.

We have referred to the question of the power required to operate the gyroscopes as being one of the disadvantages of this type of stabilizer. With a view to showing that this power was economically expended, Commander McEntee made a series of measurements of the increase in the towing resistance of the model when it was caused to roll by an harmonic rolling moment applied to the model while it was being towed through smooth water, and expressed his results in terms of the effective horsepower of the ship. From his curves it appears that at 15 knots the effective horsepower would be increased from 3,000 to 3,300 when the vessel is rolling through an arc of 25 degrees, and to 3,600 effective horsepower when the vessel is rolling through an arc of 45 degrees, corresponding to an increase of 10 and 20 per cent respectively. It is recognized that this increase in power is not absolutely the same as would be required with a vessel rolling in a seaway, but the results confirm the experience gained at sea that loss of speed occurs when a vessel is rolling heavily, and under these circumstances Commander McEntee comes to the conclusion that the power and weight devoted to the means for stabilizing a ship are more than amply repaid by the saving in the power required to drive her. We are not prepared to assert that every ship fitted with gyro-stabilizers would in all circumstances prove a cheaper ship than one not so fitted, but it is apparent from the above experiments that not only does the gyroscope yield a very effective method of stabilizing a vessel, but that a vessel thus stabilized would require less power to drive her at a stated speed than one not so fitted, and this latter fact must reduce the cost of operation and add to the earning power of the ship.—"Shipbuilding and Shipping Record," Jan. 20, 1921.

FLEXURAL BUCKLING.*

By Albert E. Guy, Member, A. O. A.

Columns, struts, and in general, all structural compression members, are designed to withstand not only the compressive stresses applied axially but also the incident tendency to give way laterally by bending, or more properly, by buckling. When the case is simple and well defined, for instance that of a strut of uniform section, the area of that section is so chosen that the stress per unit of area is well within the limits sanctioned by practice; and in addition, the least moment of inertia of that section must be such as to insure the safe elastic stability of the strut.

That beams subjected only to simple bending efforts may buckle precisely as axially loaded struts do was shown for the first time about eighteen years ago, when the results of some four thousand tests were published in the columns of the "American Machinist." A number of incidents which came recently to his attention, such as the giving way of the jib of a locomotive crane, causing the death of two men, the deformation of certain parts of ordnance materials, and some problems involving combinations of stresses, prompted the writer to give additional publicity to the subject, thus emphasizing certain points of vital importance and providing handy tables for the use of the designer.

As it was necessary to verify the effects of some combinations of stresses, the writer was duly authorized to proceed, and a very simple apparatus was quickly rigged in the writer's office, and that furnished an excellent occasion for demonstrating the fundamental laws before officers of several Ordnance Sections.

THE FOUR PRINCIPAL LAWS.

These laws are stated here as applying exclusively to cantilever beams of uniform rectangular section; it will be shown, however, that they apply equally well to *uniform* sections of any shape, and to beams held or supported in various manners.

Three axes are to be considered:

(1) The longitudinal axis of the beam; assumed to be originally straight;

(2) The greatest, or major axis of the cross section, in the direction of which the load is applied *and maintained*; this axis is always understood as in the same plane with the longitudinal axis; hence, these two axes and the direction of the load should remain in the same plane until the ultimate

*[A problem of marine engine design encountered some twenty-five years ago, and in which the question of least weight was of prime importance, led the author of this article into a merry chase after theoretical data for its solution. This data he did not find because it did not exist; then he resorted to experimentation and soon discovered the new laws of Flexural Buckling. The results of his tests were published in part in 1902. Recently several questions of design of ordnance material were brought to the author's attention, and he found that their solution depended to a large extent upon the application of the Flexural Buckling formulæ. However, it was necessary to extend and verify these latter in order to meet some particularities of these questions. Permission was given for conducting the required tests and the Ordnance Department, realizing that the resulting knowledge is likely to prove of value in general engineering work, has gladly acquiesced in the publication of this new development.

The field opened by the author is of very large extent, and only a few of the usual cases have been investigated. An immense amount of work remains to be done through experimentation, in seeking to establish the mathematical theory of Flexural Buckling. Anyone acquainted with the theory of columns as treated by Euler and other celebrated mathematicians will realize the magnitude of this task. For this reason, the author urgently requests the co-operation of elasticians and engineering colleges in this important work.—Ed.]

load increment brings about the failure of the beam through lateral buckling;

(3) The smallest, or minor axis of the cross section, always perpendicular to the "axes-load" plane.

First Law—A horizontal beam held fixedly at one end and loaded at the other will buckle when the load reaches a certain value, and with the same load acting always vertically the beam will buckle for any given angular position of its longitudinal axis, from the horizontal to the vertical; that is, through a range of 90 degrees.

Second Law—The load causing the buckling of a beam through flexure is inversely proportional to the square of the length upon which it acts directly.

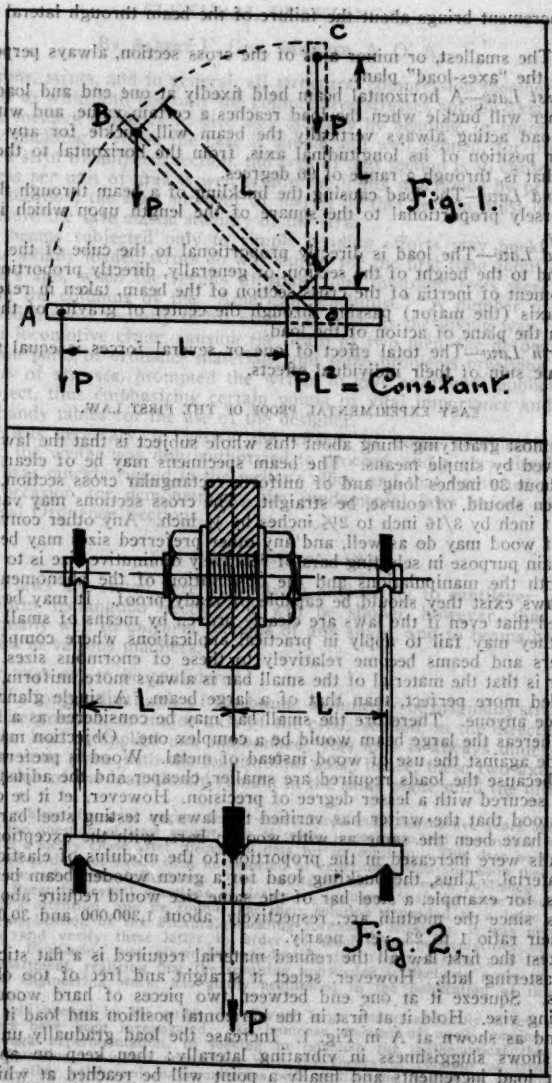
Third Law—The load is directly proportional to the cube of the thickness and to the height of the section, or generally, directly proportional to the moment of inertia of the cross section of the beam, taken in reference to an axis (the major) passing through the center of gravity of the section, in the plane of action of the load.

Fourth Law—The total effect of one or several forces is equal to the algebraic sum of their individual effects.

EASY EXPERIMENTAL PROOF OF THE FIRST LAW.

The most gratifying thing about this whole subject is that the laws may be proved by simple means. The beam specimens may be of clear white pine about 30 inches long and of uniform rectangular cross section. The specimen should, of course, be straight. The cross sections may vary between 1 inch by 3/16 inch to 2½ inches by ¾ inch. Any other convenient kind of wood may do as well, and any other preferred sizes may be used. The main purpose in selecting bars of relatively diminutive size is to facilitate both the manipulations and the observation of the phenomena. If such laws exist they should be capable of ready proof. It may be again objected that even if the laws are clearly proven by means of small specimens they may fail to apply in practical applications where compression members and beams become relatively to these of enormous sizes. The answer is that the material of the small bar is always more uniform, could be called more perfect, than that of a large beam. A single glance will convince anyone. Therefore the small bar may be considered as a simple case whereas the large beam would be a complex one. Objection may also be made against the use of wood instead of metal. Wood is preferable to metal because the loads required are smaller, cheaper and the adjustments can be secured with a lesser degree of precision. However, let it be clearly understood that the writer has verified the laws by testing steel bars; the results have been the same as with wooden bars, with the exception that the loads were increased in the proportion to the modulus of elasticity of the material. Thus, the buckling load for a given wooden beam being 17 pounds, for example, a steel bar of the same size would require about 391 pounds, since the modulii are, respectively, about 1,300,000 and 30,000,000 and their ratio 1 to 23, very nearly.

To test the first law all the refined material required is a flat stick, say one plastering lath. However, select it straight and free of too obvious defects. Squeeze it at one end between two pieces of hard wood in a swiveling vise. Hold it at first in the horizontal position and load it at the free end as shown at A in Fig. 1. Increase the load gradually until the beam shows sluggishness in vibrating laterally; then keep on applying smaller load increments and finally a point will be reached at which no further increase will be possible. The beam will then fall, or warp, to one


C
B
P
L
P
Fig. 1.
A
L
P
PL² = Constant.
L
L
P
Fig. 2.

side and would collapse entirely if not prevented by a slight pressure from the fingers applied at the loading point. Now remove the load partially, shift the beam to *any angular position B*, and apply again, but gradually, the full load. The beam will collapse under precisely the same load as at A. Shift the angular position of the beam, after partial unloading, to the vertical direction CO, and again the same total load will produce the buckling.

This test may be repeated with different lengths of the same bar, but, for each length, the load will of course be different. It will be found that buckling will occur at any angular position of the beam on the quadrant.

For the complete success of this test three things are essential: (*a*) The load must always act in the vertical direction; (*b*) the load must be so applied at the end of the beam as to leave to the latter complete freedom to bend, and to warp laterally when the limit is reached; (*c*) the direction of the load must lie in the plane passing through the major axis and the longitudinal axis.

At the beginning of previous experiments the suspension members of the load were defectuously arranged, and it was found that they actually restrained to some extent the free warping of the loaded end. The result was that the recorded loads followed broadly the general law, but with some irregularities which could not very well be accepted as matters of course. The trouble was eradicated entirely by the use of the suspension device shown in Fig. 2. This device hardly needs description. It is composed of a small steel rod, notched at each end, attached firmly to the beam by means of nuts and washers, and so disposed that its axis coincides with the minor axis of the section, and that the notches are equidistant from the major axis; two steel strips fit on knife edges in the notches and reach downwards to another notched beam, at the middle of which a knife-edged strip rests while supporting the loading platform. The whole forms a suspension parallelogram, freely deformable to suit the angular displacements of the section while insuring the direction of the load to pass through the intersection of the minor and major axes.

A little difficulty may be experienced when nearing the limit load; if the latter's direction is not kept precisely in the axial plane, the beam being considerably fleat in ordinary flexure will develop a marked tendency to warp through induced twisting, which the operator is at first apt to mistake for buckling; however, with a little experience and care, the plane of action may be re-established, and with slight additional loading the true limit will be obtained.

THE SECOND LAW.

The beams being held horizontally in the vice, with the greatest permissible length between the points of support and of loading, the buckling load is determined in the manner afore-described. The length is then reduced by a predetermined amount and the new buckling load obtained. This process is continued until the length becomes so short that the beam may break by simple bending. As a matter of precaution it is well to ascertain by a few tests what this limit is for beams of given cross sections, and then to remain well within the limit.

It will now be found that for each length tested *the product of the recorded load by the square of that length is a constant.* The approximation is so close that there is no doubt about the generality of the law, particularly so if great care has been exercised in making the tests and, of course, if the specimen used is of uniform size and structure throughout.

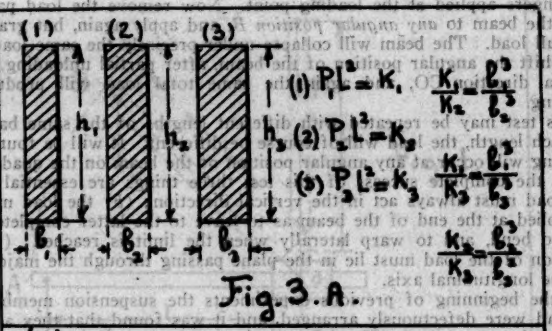


Fig. 3. A.

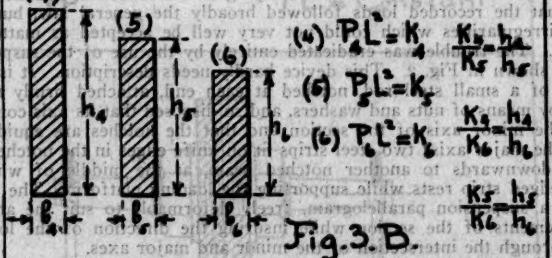


Fig. 3. B.

$\frac{K_1}{K_4} = \frac{b_1^3 h_1}{b_4^3 h_4}$ $\frac{K_1}{K_5} = \frac{b_1^3 h_1}{b_5^3 h_5}$ $\frac{K_2}{K_6} = \frac{b_2^3 h_2}{b_6^3 h_6}$, etc.

Calling K a general constant:

$PL^2 = K b^3 h$. Euler formula for a column held at one end and loaded at the other is: $PL^2 = \frac{\pi^2}{4} EI$, where I = least moment of inertia of the section $= \frac{b^3 h}{12}$.

Fig. 3. C.

This product is termed by the writer *the buckling moment of the beam.* For uniformly distributed loads and for special forms of loading the length function does not always remain, in its final state, expressed as the square of the length, as will be shown later. In view of those cases the buckling moment might be expressed more generally as the product of the load by a function of the length. This is offered here as a convenient expression and if subsequent work by other investigators should lead to a better one, the latter should be adopted if at all beneficial.

THE THIRD LAW.

Select several bars of the same kind of wood and having practically the same modulus of elasticity. Let three of these, for example, have the same uniform height of section h, but each of different uniform thickness, b, Fig. 3 A. Let three others be of the same thickness, b, but each of different height h, Fig. 3B. After each of these bars is tested for the Second Law, and the results are compared, it is found that for the three first bars the constants, and consequently the loads, are proportional to the cube of the thickness; for the three others the loads are proportional to the heights of the section. A further comparison brings out the fact, Fig. 3C, that the loads are proportional to the products of the cube of the thickness by the height. This product multiplied by a general constant K is equal in each case to the value of the buckling moment. This constant remains to be determined.

Now, since the buckling load is the same for any regular position of the beam, when the latter is truly vertical it becomes a strut and is not any more subjected to direct bending action. The law established by Euler for the buckling of a column under similar conditions of fixation and of loading is expressed by a formula which contains all the factors uncovered by the above tests, plus the modulus of elasticity of the material and one sinusoidal function. The obvious inference is that the product of these two constant factors equals the value of K, the general constant. The proof is easily made.

The modulus of elasticity is readily obtained from simple bending tests. As a matter of fact it should be determined before proceeding with the buckling experiments, when the bars are still intact, and not marred by the jaws of the support. A specimen is placed horizontally flat upon two knife-edged supports, their distance apart being carefully measured. It is loaded at the middle just enough to make it quite rigid, but no record need be made of the load. The vertical distance from a point at the middle of the beam to a horizontal datum plane is very carefully measured. Then a known weight is added to the first load and the corresponding deflection is measured. Knowing the length between supports, the dimensions of the cross section, the second weight applied and its corresponding deflection, the modulus E is calculated by the well known flexure equation.

Introducing the value of the calculated constant in the equations derived from the experiments, it is found true to a most convincing degree of approximation. At least, that has been the experience of the writer when dealing with a large number of tests.

THE FOURTH LAW.

With the beam placed horizontally and fixed at one end, as before, one load is applied at the other end and another at some intermediate point of the length, Fig. 4. Either one may be increased until buckling occurs. It is found that each bending moment multiplied by its own lever arm may

be considered as a component buckling moment. In other words, P×L squared, being the buckling moment for one single load, equals $P_1 \times L_1$ squared plus $P_2 \times L_2$ squared, and for any number of bending forces acting vertically along the beam the resultant buckling moment equals the sum of all the component buckling moments. This is a most important result.

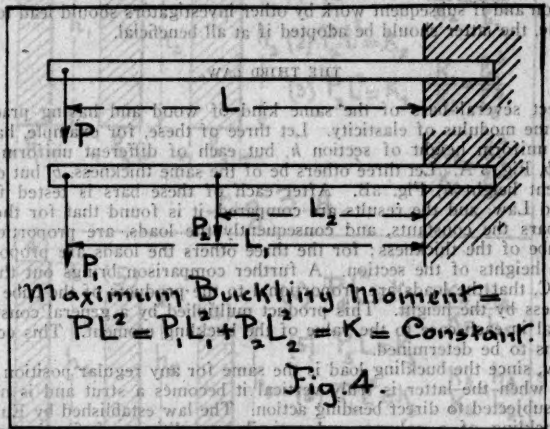


Fig. 4.

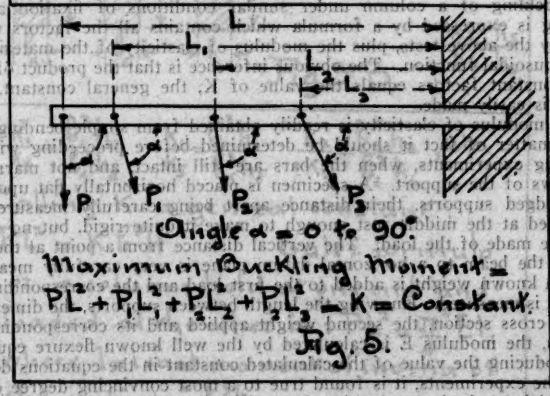


Fig. 5.

This result may be extended in a most useful manner by taking the first law into account. Thus, when any number of forces act on the beam, as shown in Fig. 5,, whether perpendicularly or obliquely to its longitudinal axis, but always along the action plane passing through the major axis of the section, the resultant or total buckling moment is the sum of all the component buckling moments, irrespective of the angle of direction of the forces, provided, however, that this angle, as shown is equal to or less than 90 degrees.

If the directions of the forces are not all along the plane of action, the above does not hold and the case becomes very complex because the beam is then subjected to twisting moments which are bound to cause local warpings and to weaken considerably the resistance of the beam to plain buckling. This question requires careful investigation and unfortunately does not seem susceptible of easy solution, if at all.

When the angle of direction of any one force is greater than 90 degrees the expression of its buckling moment takes a special form, which includes a function of that angle. As will be seen later on, this function comes in also for the smaller angles, but it vanishes automatically.

COMPOSITION OF FORCES VS. COMBINATION OF EFFECTS OF FORCES.

Fundamentally, one force acting upon one material point in space may be resolved into at least two components. The simultaneous actions of these components upon the point are equivalent to the single action of the original force. The same holds true of the composition of velocities. Very simple tests, however, bring forth a most interesting paradox, for which a satisfactory explanation has not yet been found.

In Fig. 6 a beam is shown held obliquely. At the free end a suspension parallelogram is adjusted and connected to a loading platform A. On each side of the beam one wire is connected to the cross bar at the free end, and these running over two small pulleys at the other end, extend downward for connection to a second loading platform B. Any load placed upon this platform produces a true axial compression of the beam. Any load on A bends the beam, and when sufficiently increased buckles it. With each platform loaded the direction of the pull of the wires ceases to be truly axial, but because of the first law the effect of the load B is not altered; that would complicate the following demonstration, however, and for the purpose in view the curvature due to bending will not be considered here, the beam being assumed to remain straight.

If the buckling load P for the constant length L is ascertained according to the first law, it may be applied indifferently either at A or at B. The tests show that to be the case. If part of that load is applied on one platform it is also found that the load applied on the other platform and which finally brings out the buckling is precisely the difference between P and the first load. This demonstrates that, experimentally, and *for this particular mode of action, the effects of forces add.*

Assuming that the beam is placed at an angle of 60 degrees, and merely for convenience, that the buckling load is 41 pounds. If this force is resolved into two components, in strict accordance with usual practice, there will be one bending force of 20.5 pounds and one axial compressive force of 35.5 pounds, very nearly. Ordinarily it would be taken for granted that the effect on the beam would be the same whether the single force of 41 pounds is applied vertically or the two components be applied simultaneously at the same point of the beam. These tests show that that is not the case; on the contrary, if a load of 35.5 pounds is placed on B not more than 5.5 pounds can be applied on A in order to produce buckling. Moreover, the load at A may be truly vertical, or oblique, or again be normal to the longitudinal axis and in effect a true bending force acting along the direction of the bending component.

The obvious conclusion from these observations is that—in so far as buckling is concerned—it is incorrect and unsafe to replace one force by its components when the resistance thereto is a combination of molecular actions within the material of the beam.

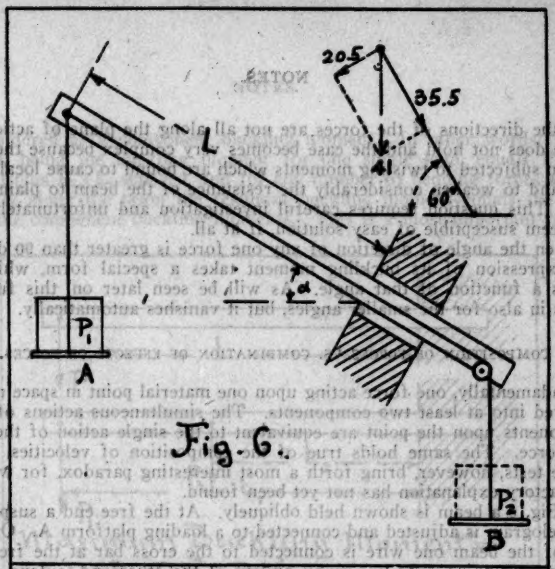
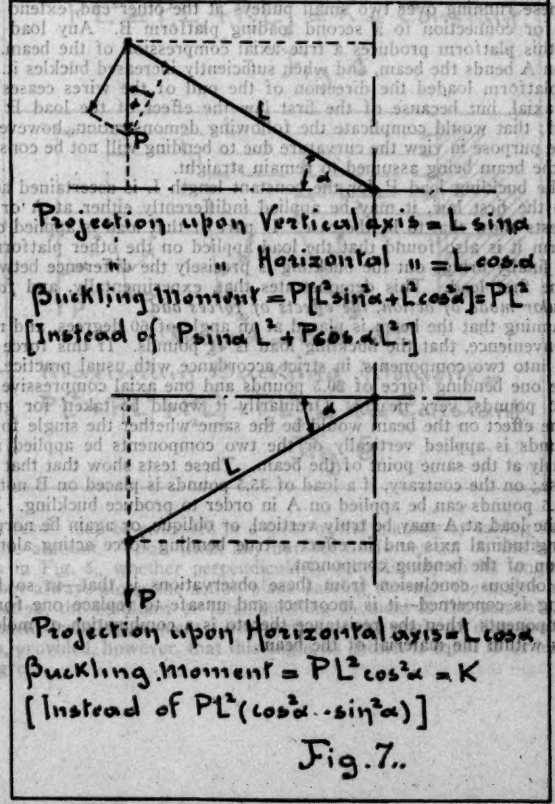


Fig. 6.



Projection upon Vertical axis = $L \sin\alpha$

" Horizontal " = $L \cos\alpha$

Buckling Moment = $P[L^2\sin^2\alpha + L^2\cos^2\alpha] = PL^2$

[Instead of $P\sin\alpha L^2 + P\cos\alpha L^2$]

Projection upon Horizontal axis = $L\cos\alpha$

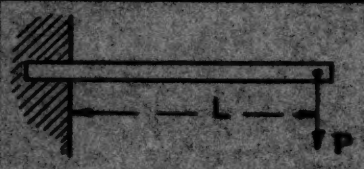
Buckling Moment = $PL^2\cos^2\alpha = K$

[Instead of $PL^2(\cos^2\alpha - \sin^2\alpha)$]

Fig. 7.

β = Buckling Moment for Beams of uniform Section throughout.
E = Modulus of Elasticity. I = Moment of Inertia of cross section referred to axis in plane of action of load. $\pi = 3.1416$.

1

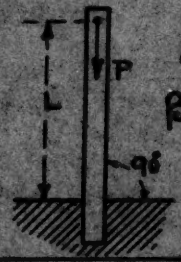


$\beta = PL^2 = \frac{\pi^2}{4} EI.$

2

$\beta = PL^2 = \frac{\pi^2}{4} EI$

3



Axial load

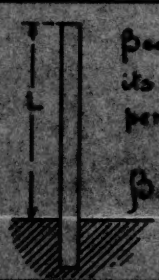
$\beta = PL^2 = \frac{\pi^2}{4} EI.$

4

Several axial loads.

$\beta = PL^2 + P_1L_1^2 + P_2L_2^2 + \ldots = \frac{\pi^2}{4} EI$

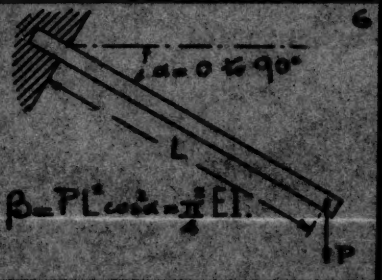
5



Beam sustaining its own weight p per unit of length.

$\beta = pL^3 = \frac{2}{\pi^2} EI$

6



$\beta = PL^2 \cos\alpha = \frac{\pi^2}{4} EI.$

7

8

9

7
$\beta = PL^2 + P_1L_1^2 = \frac{\pi^2}{4}EI$
8
$\alpha = 0$ to $90°$
$\beta = PL^2 + PL_1^2 + PL_2^2 = \frac{\pi^2}{4}EI$
9
Uniform load p per unit length.
$\beta = pL^3 = \frac{\pi^3}{4}EI.$
10
Uniform load p per unit length.
$\beta = \frac{pL^3}{3} + P_1L_1^2 = \frac{\pi^2}{4}EI.$
11
Uniformly distributed load p per unit length.
$\beta = \frac{pL^3}{3} + PL^2 + PL_2^2 + PL_3^2 = \frac{\pi^2}{4}EI.$
12
$\beta = PL^2 = \pi^2EI.$
13
$\beta = PL_1L_2 = \frac{\pi^2}{4}EI.$
14
$\beta = PLL_1 = \frac{\pi^2}{4}EI.$
15
Uniform load p per unit length.
$\beta = p\frac{L^3}{2} = \pi^2EI$

For instance, assume that a vertical load of 41 pounds be a safe value for a given beam, the sum of the components, for a 60-degree angle, would be 56 pounds and the buckling moment 36.6 per cent greater than with the single vertical load, if the components were substituted for the latter.

Again, for the same beam, 41 pounds being the ultimate load, assume 30 pounds as a safe vertical load. If this force is replaced by its two components, 15 and 26 pounds, their sum is 41 and their simultaneous action would buckle the beams, whereas 30 pounds would not.

COMPLEMENT TO THE FIRST LAW.

The finding that the effects of the forces are additive instead of combinative, as expected, makes it awkward to understand and to explain the resistance of the beam held inclined above the horizontal and still more so when inclined below. For the upper condition, the simultaneous effects of the components would give, as per the equation in Fig. 7, the same buckling moment for two points only, when the angle equals zero or 90 degrees. For any intermediate angle the moment would be greater and would be maximum at 45 degrees with a value then 41.4 per cent greater than the value recorded. The slight discrepancies observed in the tests were fully within admissible limits, and the buckling loads practically identical for any angular position.

It therefore seems justifiable to consider the buckling moment for the beam in the upper position as composed of two simultaneous moments, one affecting the projections of L upon the vertical axis and equal to the product of the load P by the square of that projection, and another equal to the product of the load by the square of the projection of L upon the horizontal. This is equivalent to treating the vertical projection as a strut under axial compression and the horizontal as a beam subjected to simple bending. Each effect tends towards buckling the inclined beam, consequently the two may add. Since the expressions of the moments are functions of the angle, the sum of the squares of the projections reduce these functions to unity and they consequently disappear from the equation.

With the under position it is seen that the vertical moment does not tend towards buckling the beam, but, on the contrary, seemingly to steady it; it may therefore be omitted. The horizontal moment remains, and alone tends to produce buckling. Consequently the first law may be extended by an exception to the rule, and expressing that the buckling moment of a beam held inclined below the horizontal and loaded vertically at the free end equals the product of the load by the square of the projection of the length upon the horizontal.

Originally the writer was led, from the results of a few tests, to the conclusion that the vertical moment was to be subtracted from the horizontal, since it seemed to have a steadying effect; more recent tests have shown conclusively that it should be omitted. In other words, only the effects tending towards the same end should be added. The loads applied to the beam in the under position increased very rapidly with the angle; for instance, at 45 degrees, the load is twice that at zero, for the same length, of course. Besides, while the angle may be very carefully adjusted at the point of fixation, under such considerable loads the curvature of the beam is very pronounced, so that at the point of application the angle may be 5 to 10 degrees greater. The tendency of the beam to twist at several points of its length makes it very difficult to secure accurate results, and this explains the uncertain conclusions derived from the original tests.

DATA SHEET OF BUCKLING MOMENTS.

On a separate plate are shown fifteen cases of loading, and the formula for the ultimum buckling moment is given for each. The formulae expressing the effects of one or of several loads are derived from direct experiments, those dealing with uniformly distributed loads have been established by computation. For practical use a factor of safety must be introduced; its value depends on the mode of loading and of support. However, it would be safe for all practical applications to multiply the buckling moments by 2. Effectively, the tests show that, as the load increases gradually, the beam vibrates very vigorously laterally at half load, and it is only after the three-quarter load is passed that the vibrations become decidedly sluggish.

Throughout this article only beams of uniform rectangular cross-section have been dealt with, and the impression might obtain that the formulae given here do not apply to other forms of sections. It should be stated, however, that they apply equally well to any usual form. Numerous tests were made with T, I and U sections and the results were invariably expressed in function of the moment of inertia of the cross-section, referred to its vertical axis.

In a following article the great importance of the beam supports, of the attachments of the loads, and of the secondary moment induced thereby will be indicated for a few cases selected at random. The matter covers such a wide field, heretofore unexplored, it seems, that it is deemed proper to urgently request the co-operation of the engineering colleges in the pursuit of laboratory investigations, which are bound to prove useful mainly in two ways: in the elucidation of the elasticity behavior of materials and, for students principally, in a facile and true visualization and understanding of the application and effects of stresses.—"Army Ordnance," March-April, 1921.

OBITUARY.

REAR ADMIRAL FRANK HARVEY BAILEY, U. S. N.

In these days of constant progress in engineering the death of a man who has retired from active service for nearly a decade is apt to pass without sufficient notice of the good work he has done unless some friend who was a junior associate during his active career contributes a few memoranda as a tribute of respect and affection. The writer's personal acquaintance with Admiral Bailey covered almost forty years, and included a short cruise and an extended tour of duty in the Bureau of Engineering, during which times the association was constant and intimate.

Admiral Bailey was one of the most modest of men, with the natural result that his great ability and lovable character are only known to those who were thrown with him intimately. He gave his whole attention to his work and none to exploiting what he had done, so that the general public hardly knew him at all. He was born in Cranesville, Pa., in June, 1851, but moved to Gowanda, New York, in 1868. He took a course in Scio College, Scio, Ohio, and then, in 1873, entered the Naval Academy as a cadet engineer in the third class of those who took the two-year course. His remarkable ability and aptitude for engineering showed from the beginning and he graduated in June, 1875, at the head of a class which contained a number of very able men. From that time until his retirement in 1913 he performed the usual alternation of duty at sea or on shore, making a record which was always distinguished for the highest efficiency.

He was Professor of Marine Engineering in Cornell University by detail from the Navy from 1885 to 1888, where

he enjoyed the respect of the faculty and students for his attainments in his own branch of the profession and his general proficiency as an engineer. From 1891 to 1896 he was on duty in the Bureau of Engineering, most of the time as Chief Designer, and it was during this period, and a later similar assignment, that his very best work was done for the Navy.

Admiral Bailey was a born engineer and his mind worked like a beautiful piece of machinery in perfect condition. Every problem that came up for solution was given careful and thorough study, and when he had finished, the work was done and satisfied everybody as the best solution that could be made. He possessed in the highest degree that splendid common sense which another brilliant engineer once stated to be nine-tenths of good engineering. One of the most notable instances of this trait was his rehabilitation of the air pump engines of some nine or ten of the early ships of the new Navy. A description in detail of his analysis and solution of this problem would be very interesting, but perhaps out of place here. Suffice it to say that he made a complete success and secured entirely satisfactory operation, while all the other designers failed to improve the original condition which was a constant source of worry. Many other instances might be given to show the clearness with which his mind handled these problems, but this one was a simple and very striking illustration.

Admiral Bailey was one of the most lovable of men on account of his remarkable evenness of temper and uniform kindly sympathy. It seemed impossible for any amount of nagging to arouse his anger. He only smiled and turned the incident aside as one of the unpleasant things of life which have to be borne, but which the wise man does not permit to disturb his serenity. It is safe to say that he had no enemies and that all of us who were privileged to enjoy his friendship will always count it as one of the precious possessions of our lives.

He was devoted to the Navy and took a particular interest in its efficiency as a whole, not content to see his own branch doing good work, but anxious that the whole great organization as the marine arm of the military power should work harmoniously, and be pervaded by a spirit of enthusiasm in all its officers. His study of conditions as they then were led him to suggest as early as 1890 the amalgamation of the line and engineer corps which finally came about in 1899. The writer remembers well numerous occasions on which Bailey discussed this point with him in all its bearings. It is interesting in this connection to note that another talented officer of great ability who has recently retired after a distinguished career as one of the best strategists in the service also discussed this same subject with the writer about 1896 and suggested amalgamation from the standpoint of a line officer. Thus it happened when the late Admiral Evans proposed the amalgamation to the Personnel Board of which the writer was a member, the latter was in a position to say that he had discussed this with some of the ablest officers in both the line and engineer corps and knew that they were favorable to it. The idea, as is known to students of the subject, dated back as far as the time when Secretary Gideon Wells suggested the same thing in 1865, but the service was not then ready for it. As President Roosevelt (then Assistant Secretary of the Navy and Chairman of the Personnel Board in 1897) so well expressed it: "We are not making a revolution; we are merely recognizing and giving shape to an evolution, which has come slowly but surely and naturally, and we propose to re-organize the Navy along the lines indicated by the course of the evolution itself."

Admiral Bailey took part in the two wars which occurred during his mature life. He was Chief Engineer of the *Raleigh* at the Battle of Manila, which was the first naval battle of any importance in which our vessels had engaged since the Civil War. The writer was on duty in the Navy

Department at that time and a few days after the battle received a letter from Admiral Bailey written from Hong Kong in the early part of April, 1898. In this letter he said: "It looks very much as if we would have to lick the Spaniards at Manila, but we are not worrying about it and believe that we shall have a comparatively easy time."

During the World War he was recalled from the retired list and assigned to duty in the Design Division of the Bureau of Engineering, where his ripe experience and splendid judgment enabled him to render valuable service.

Thus our friend has made his record and passes on as one of that body of fine and faithful men who have ever maintained the high reputation of the Navy by performing faithfully and efficiently every duty as it came along without thought of special reward or as to whether it might make them famous. Many names will occur to all who are familiar with the history of the service as worthy of inclusion in this faithful band. The record of their lives is an inspiration to the younger men in whose keeping is the reputation of the Navy today; and it is a great satisfaction to all of us who love the service to feel that the same high standard of devotion to duty which animated our departed friend and the others like him permeates the service today and assures the maintenance of the highest efficiency.

WALTER M. MCFARLAND.

BOOK REVIEWS.

A HANDBOOK OF PRACTICAL SHIPBUILDING. BY J. D. McBRIDE. Second Edition, Revised and Enlarged. D. VAN NOSTRAND COMPANY, 8 Warren St., New York. 356 pp. Ill. $3.00 net.

This handbook, as its name implies, is for the practical shipbuilder, the man working in the shipbuilding yard, to give him a general knowledge of the way a ship is built from the start of work in the mold loft to the time of completion and dock trials. Further, it gives in more detail a description of the actual work of the different trades in the shipyard, with particular reference to the work of the shipfitters and their allied trades, mold loftsmen, shipfitters, chippers and calkers, and riveters. For these trades the methods of work are carefully given with descriptions of tools, their proper use and care.

The book is prepared as applying specifically to yards building merchant cargo ships, and particularly, it would appear, to those yards which grew up during the war in which extensive foundries, machine shops, etc., were not included. Merchant practice is consequently almost entirely referred to in specific instances, but the general information is nevertheless of value for any shipyard.

In no sense is this a designer's handbook, and to this extent it may be said to fill an existing want, for while there are many handbooks for designing naval architects and engineers, this is, as far as is known, the only handbook intended primarily for the man in the shipyard. Nevertheless it is a book which may well be studied by an apprentice ship draftsman or any engineering draftsman taking up ship work, as it brings out the problems of the steel worker in the yard and the effect design may have upon the ease or difficulty of this work.

In this second edition, the original edition has been enlarged by the addition of two chapters, one on stop waters and oil stops and one on tank testing. These are two most important features in building a ship from the shipfitter's point of view, and in fact from any point of view. Stop waters—and particularly oil stops—are a subject which have come specially to the fore with the carrying of oil for fuel, and a proper understanding of their function together with a knowledge of what materials to use and how to use them is of great importance in obtaining eventual water and oil-tightness of the various parts of the ship. With regard to tank testing, this is one of the elements of ship construction in which skill makes itself of the utmost value, and in which skill, greater or less, may result in large savings or expenditures. The information given in this last-named chapter should therefore be of great value to the men in a testing gang. In this chapter it is stated that "for a man to be a thorough tank tester he must have a good knowledge of not only *what* he is doing, but *how* to do it correctly and *why* he is doing it." It is believed this sentence epitomizes the idea with which the whole handbook is written.

The book is simple in language and easy to read and understand. It should not only be a valuable textbook for trade schools in shipyards and apprentice ship draftsmen particularly under conditions which have prevailed during the past few years, but should be also a good book of reference for supervisors in the shipyard and for those men of the steel workers trades who wish to advance by a thorough knowledge of their profession.—H. S. H.

MODERN MARINE ENGINEERING. PART I. THE FIRE ROOM. BY HARRY G. CISIN. D. VAN NOSTRAND CO. 205 pp. Ill. $3.00 net.

This book has been prepared with a view to its use as a school and college textbook, as well as for the practical man

who desires to broaden his knowledge. It is divided into a series of short lessons with questions at the end of each lesson to bring out the various points discussed. The book is well arranged and describes in easily understood language the construction, installation and operation of boilers and fire room auxiliaries. There are a number of good illustrations.

There are several statements made in the book with which the reviewer cannot agree. These are:

Page 39: "In all express boilers high horsepower is obtained at a sacrifice of fuel as compared to the fire tube boilers."

Page 40: "For highest efficiency per sq. ft. of heating surface—fire tube."

Page 88: "Main stop valve—casting of—cast iron." (The best practice demands that valves or fittings for high pressure steam must not be made of cast iron—cast steel is suitable and is largely used.)

Page 99: "Coal buckets—carry about 35 lbs. of coal." (Coal buckets should hold about 115 lbs. of coal.)

Page 117: "(C) Use of pure mineral oil is a preventive measure." (Evidently means "Use of pure mineral oils *for lubricating purposes* is a preventive measure.)

Page 121: "The closed fire room system—is the least economical of all forced draft systems, is expensive to install and is hard on the firemen." (In the opinion of the reviewer this system is the most efficient system and once installed requires less repairs to keep in efficient condition. This system overcomes the ever present menace of serious injury to personnel existing in the closed ashpit system due to fire blowing back into the fire room from the excess air pressure in the furnace.)—F. J. C.

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